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INSTITUTE OF MARINE ENGINEERS,
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Patron: HIS MAJESTY THE KING.

LORD WEIR OF EASTWOOD, President– 1920/21.
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ENGINEER VICE-ADMIRAL SIR GEORGE G. GOODWIN, R.N., K.C,B., LL.D., President—1922/23.

PAPERS ON

INTERNAL COMBUSTION ENGINES.

182299.
9.7.23.
M.E. 638.

READ AND DISCUSSED
AT
The Premises of the Institute of Marine Engineers, 85/8, The Minories,
Tower Hill, London, E.1.

December, 1922.

PUBLISHED BY
The Institute of Marine Engineers, 85/8, The Minories, London, E.1.

Printed by Wilson & Whitworth, Ltd.,
— Broadway, Stratford, E.15. —

PREFACE.

The heavy oil engine was brought forth at a time when there were abundant mechanical resources for its development and in consequence there has been rapid progress in design. One of the results of this quick evolution has been that this type of engine has gone ahead of popular ideas and is intimately known to only a comparatively small section of marine engineers.

The desire to rectify this state of affairs led the Council of the Institute of Marine Engineers to organise and publish the series of lectures which comprise this volume; the need for such a book seemed to be pronounced, for although many valuable works on the theory and design of internal combustion engines exist, it was considered that the experiences of those who had been in charge of them in the experimental stage and in service would be of great value to those who desired a closer acquaintance with this new form of prime mover.

The Council accept no responsibility for the opinions expressed in the following papers, the diversity of which is eloquent of the need for further test, so that the best form may be decided upon, but they desire to offer the thanks of the Institute of Marine Engineers to those who have, in the midst of other exacting duties, found time and care to write these papers, which, with the discussions and notes thereon are now put before engineers, and the marine engineer in particular, in the hope and belief that the effort will prove to have been well worth while.

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*(*Read before Lloyd's Register Staff Association and included by permission*).

SESSION 1920-21.

President: LORD WEIR OF EASTWOOD.

Oils for Internal Combustion Engines

BY MR. J. L. CHALONER.

Read Tuesday, November 16, 1920.

OPENING REMARKS.

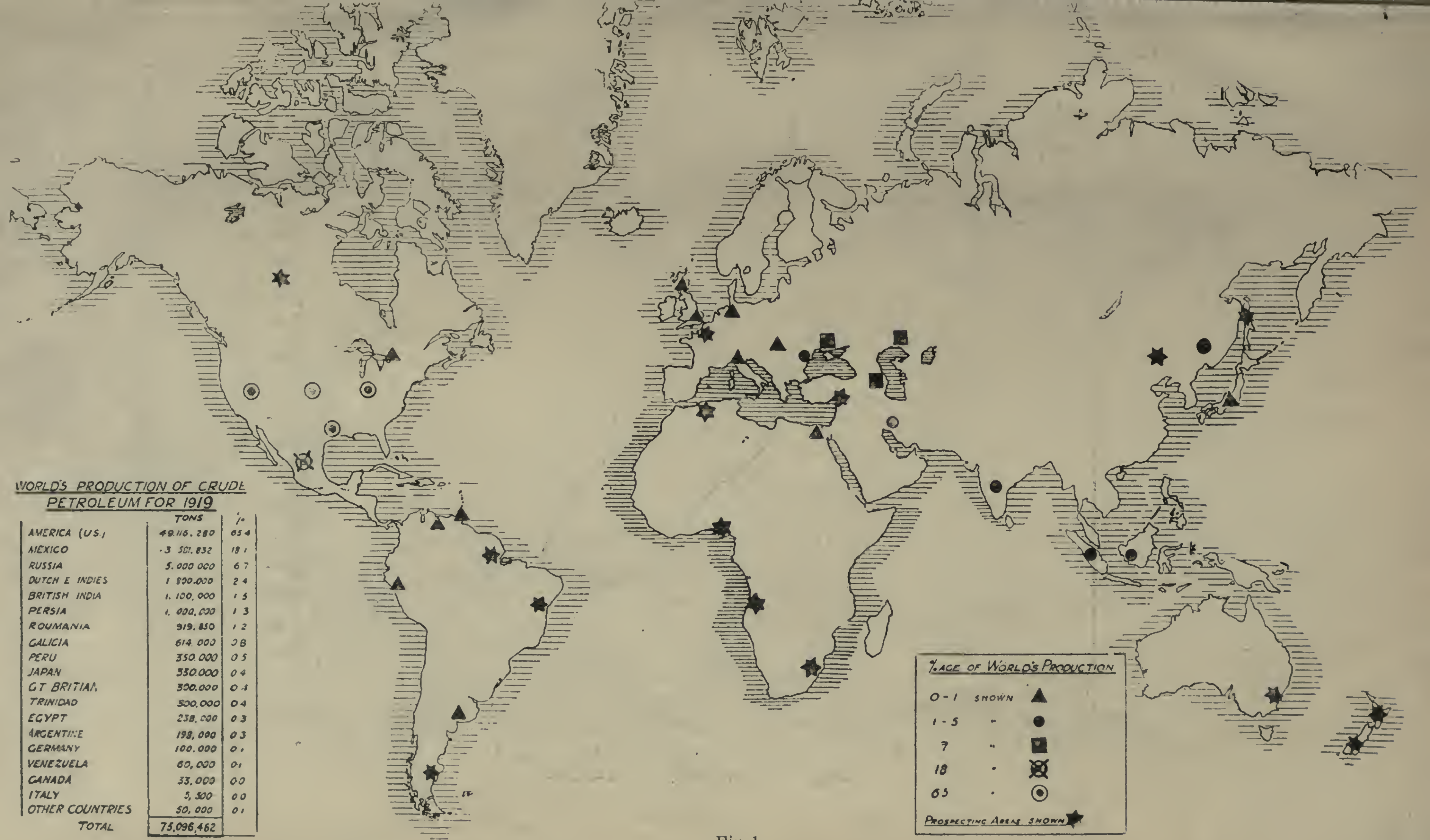
The CHAIRMAN: The Council, in arranging this series of lectures, are endeavouring specially to deal with the latest progress of the application of the oil engine,—or the internal combustion engine,—to marine purposes. The marine engineer is a little bit troubled, especially those like myself and others who have been brought up with the steam engine. It seems difficult at first sight to realise that it is possible to take the furnaces out of the boiler, put them into the cylinder, then fasten on the cover and expect the engine to work. With the marine engineer rests almost entirely the responsibility of keeping the engine turning, and that responsibility means that in the event of something going a little astray he must be able to detect the cause and remove it. Most of us have had considerable experience with the troubles of the steam engine, but the troubles of the oil engine are not so generally well-known. The Committee appointed by the Council felt that a special series of lectures should be arranged, the main object of which was not to air special grievances or special experiences, but rather to turn to the practical side, the working of the engine, and, if possible, to lay stress on those points that are likely to give trouble, so that those on watch will have an opportunity of giving consideration to these points, and learn how to put them right. More important will be the fact that with this knowledge they will be led to the anticipation of these troubles, and thus will be able to prevent them arising. That, perhaps, is the ultimate goal. The Committee thought it would perhaps help if one began at the beginning of the subject, and thus we should give first attention to the fuel, and then to chase the fuel through the engine to the exhaust. What are the troubles of getting suitable oil? How will you test a sample? How will you make up your mind as to the relative advantages of an oil out of two or three grades that may be offered you? These are some of the points for consideration, and in the second paper to be read this evening I anticipate there will be given some good suggestions on these points of selection and final determination of the oil you will purchase. Another trouble is storage. You may have a good thing and make it worse by storing it improperly. Next there is the pipe system between the bunkers and the engine. Following that is the principles of combustion, the problem of efficient combustion. We hope by the end of the session our members will have obtained a very great and substantial help in their knowledge of the working of the internal combustion engine. It is a privilege to start a new thing going, and it is hoped that at the end of the session we may be able to look back upon a valuable addition to our Transactions.

I would like to invite everybody present to take part in the discussion, for the Author will, I am sure, do all that he can to help you out of any difficulties. Some of the simpler tests will be made before us, and thus afford us a good opportunity of increasing our knowledge of the essential points. I would like to suggest to you that you take full advantage of this opportunity of making yourselves acquainted with the peculiarities of the oil engine, so that if need arises and the opportunity offers, you will be able to accept an opportunity of serving in a ship fitted with internal combustion engines.

MR. J. L. CHALONER.

Oil fuel is practically found all over the world and the world's map in Figure No. 1 illustrates the position of present and prospective oil fields. The curve in Figure No. 2 shows the development of the world's production of petroleum, from which it will be noted that whereas progress has been very rapid the total available production of oil to-day is only 3 per cent. of the total production of coal.

Whilst in former days it was never the practice to use crude oil very extensively for industrial purposes, yet the commercial and economical position did not warrant any detailed splitting up of the crude oil into its various products. To-day, however,



WORLD'S PRODUCTION OF CRUDE PETROLEUM FOR 1919

	TONS	%
AMERICA (U.S.)	49.116.280	65 4
MEXICO	·3 581.832	18 1
RUSSIA	5.000 000	6 7
DUTCH E INDIES	1 800.000	2 4
BRITISH INDIA	1. 100, 000	1 5
PERSIA	1. 000, 000	1 3
ROUMANIA	919. 850	1 2
GALICIA	614. 000	0 8
PERU	350. 000	0 5
JAPAN	330.000	0 4
GT BRITIAN	300.000	0 4
TRINIDAD	300.000	0 4
EGYPT	238, 000	0 3
ARGENTINE	198, 000	0 3
GERMANY	100.000	0 1
VENEZUELA	60, 000	0 1
CANADA	33, 000	0 0
ITALY	5, 500	0 0
OTHER COUNTRIES	50, 000	0 1
TOTAL	75,096,462	

Fig. 1.

CURVE No. 2.

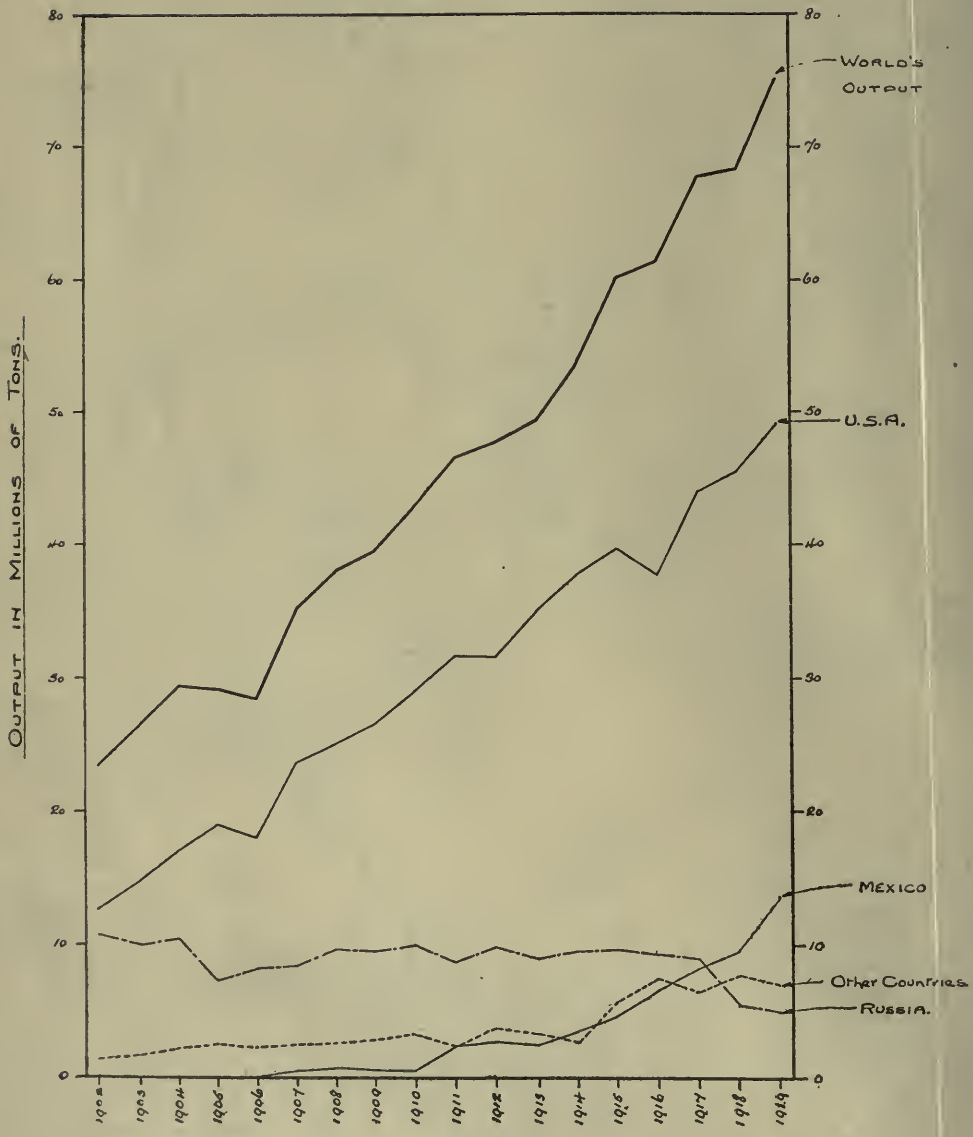


Fig. 2.

during the refining process of petroleum, the following yields of the various products are obtainable:—

Petrol	26%
Kerosene	10%
Gas Oil	5%
Lubricating Oil	6%
Fuel Oil	40%
Residue (asphalt wax)	12%
Loss	1%

Needless to say the percentage of the different yields varies considerably with the origin of the oil fuel, but for general purposes the above data represent fairly average records. (See Figure 3).

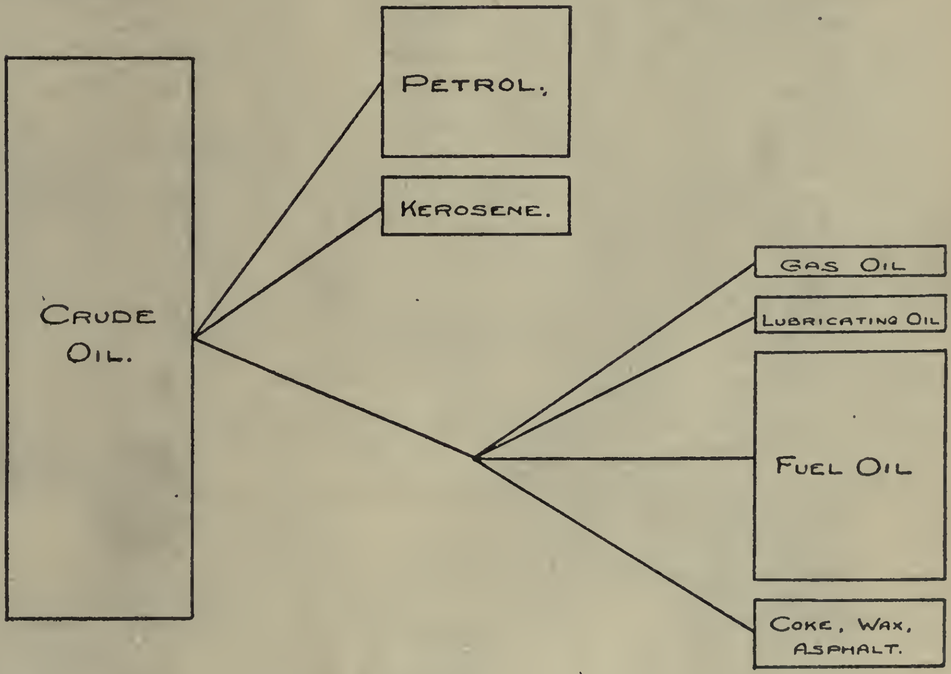


Fig. 3.

Modern engineering has found a specific function for each grade of distillate or residuum, and the general tendency is towards using the cheapest class of fuel in each case, provided, of course, that at the same time a high degree of reliability is retained together with an average maximum efficiency. As is indicated on the world's map, oil is found in practically all parts of the world, and it may be anticipated that there must be a fundamental difference in the constitution of the oil which will have an important bearing on the suitability of the various fuels. Broadly speaking, the oils found in the world may be one of three bases, viz., Naphthene base, Paraffin base, and Asphaltic base. The Naphthene bases are generally confined to the Eastern and Mid-European countries, the Paraffin bases are usually obtained from America, whilst the Asphaltic bases are mainly obtained from the Mexican and South American oil fields. To-day's refined products will, therefore, be of one of three above bases, and the suitability generally will be co-relative to the base of origin.

Whilst from a technological point of view the determination of a complete specification is a highly scientific procedure, the commercial specification confines itself to the determination of such physical and chemical properties which have been considered as a sufficient guide in the selection of a suitable fuel. It is customary to refer to oil fuels as white and black oils, in which case Petrol and Kerosene are known as white oils against Gas and Fuel Oils which fall under the black oil category.

For the purpose of indicating the scope of the investigations which may be carried out in conjunction with the determination of the physical and chemical properties of oils the following table has been compiled :—

<table>
<tr><th colspan="2">Physical Properties,</th><th colspan="2">Chemical Properties.</th></tr>
<tr><th>White Oils.</th><th>Black Oils.</th><th>White Oils.</th><th>Black Oils.</th></tr>
<tr><td colspan="2" rowspan="2">Specific Gravity.
Specific Heat.
Vapour Tension.
Flash Point.
Burning Point.
Ignition Point.
Viscosity.
Fractionation.
Calorific Value.
Oil Gas Formation.
Heat of Vapourisation.
Mechanical Impurities.
Co-efficient of Expansion.
Setting Point.</td><td colspan="2">Ultimate Analysis (Carbon, Hydrogen, Sulphur, Oxygen and Nitrogen).</td></tr>
<tr><td>Litharge Test.
Copper Test.
Iodine Value.
Unsaturated.
Compounds.</td><td>Water Content.
Ash Content.
Coke Content.
Asphalt Content.</td></tr>
</table>

From a commercial point of view and of particular interest to the Marine Engineer, however, is that specification which will make possible a quick determination as to the suitability or unsuitability of any particular fuel for any specific purpose. In connection with commercial specifications therefore, attention is confined to the following properties:—

WHITE OILS.

(*a*) *Physical Properties.*
- Specific Gravity.
- Calorific Value.
- Flash Point.
- Viscosity.
- Fractionation.
- Vapour Tension.
- Water Content.

BLACK OILS.

(*a*) *Physical Properties.*
- Specific Gravity.
- Calorific Value.
- Flash Point.
- Viscosity.
- Mechanical Impurities.
- Water Content.

(*b*) *Chemical Properties.*
- Coke and Asphalt Content.
- Sulphur Content.
- Ash Content.

I. *Physical Properties.*—1. *Specific Gravity.*—The specific gravity has hitherto always been considered as a definite guide with regard to the suitability of a fuel. Whilst such an assumption was correct when comparatively few oil fields in the world were in operation, it is becoming more and more evident that the specific gravity by itself forms no criterion for an oil. This property is, however, useful for converting tons into gallons, and in this connection it is used very frequently.

Whilst in this country the specific gravity unit is employed, in the United States the Baumé scale is used to a far greater extent. The following formula is quoted to convert the one standard into the other and *vice versa*:—

$$\text{Specific Gravity} = \frac{140}{130 + \text{Baumé reading}}$$

$$\text{Baumé} = \frac{140}{\text{Specific Gravity}} - 130$$

As will become apparent, the gravity depends on the temperature and it is customary to specify the gravity at 60°F. or 15°C. Whenever the gravity is determined at any other temperature the following adjustment should be made to obtain the approximate specific gravity at the standard normal temperature:—

Temperature Correction.	Specific Gravity Range. From ·62 (White Oils)	to	·97 (Black Oils)
For every degree above 60°F. subtract	·0001	to	·0005
For every degree below 60°F. add …	·0001	to	·0005

2. *Calorific Value.*—The Calorific value has an important bearing on consumption in internal combustion engines and this value is therefore of considerable importance to the Marine Engineer when drawing up his heat balance sheet. The average value of the various constituents of oil fuel are as follows:—

	Gross Calorific Value. per lb. of fuel.	Actual Calorific Value. per gallon.
Petrol	21,000	142,800
Kerosene ...	20,050	160,400
Gas Oil	19,500	169,700
Light Fuel Oil ...	19,100	173,900
Heavy Fuel Oil	18,750	178,200

The specification usually quotes the gross calorific value, as that is the value which is determined when testing the fuel for its heating value. From a consumption point of view, however, the net calorific value is of greater importance and the general rule for finding the net calorific value from the gross is by using the following formula:—

$$\text{N.C.V.} = \text{G.C.V.} - (97\ \text{H} + 173\ \text{H}_2\text{O})$$

where H is the percentage of Hydrogen, and
H_2O the percentage of water.

The following table gives the approximate hydrogen contents in the various distillates and residuums:—

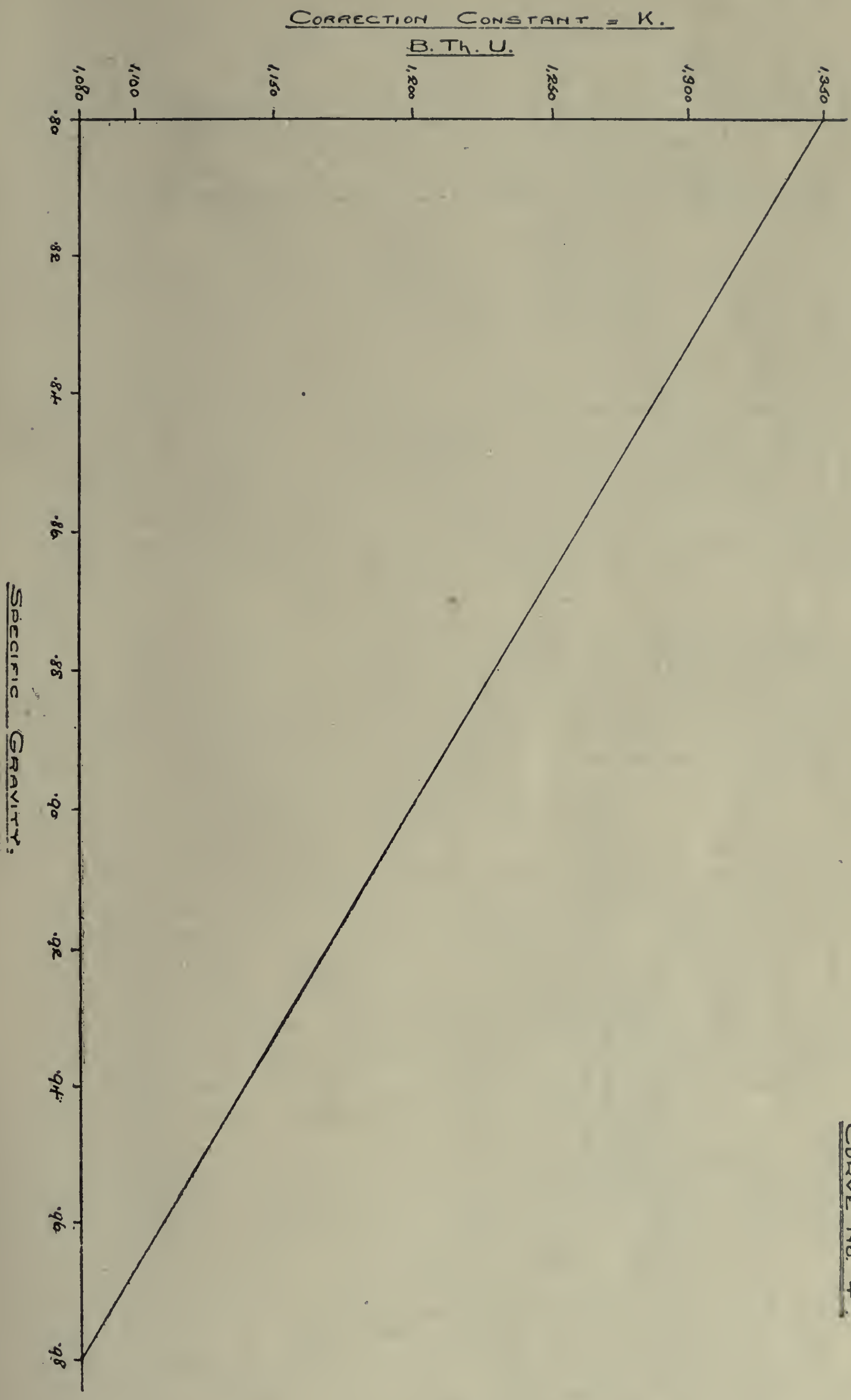
Petrol	14·5% of Hydrogen.
Kerosene	14·2% ,, ,,
Gas Oil	12·7% ,, ,,
Fuel Oil	12·4% ,, ,,

It is not always possible to determine the hydrogen contents, and the net calorific value in such a case can be determined approximately by the following formula:—

$$\text{N.C.V.} = \text{G.C.V.} - (\text{K} + 173\ \text{H}_2\text{O})$$

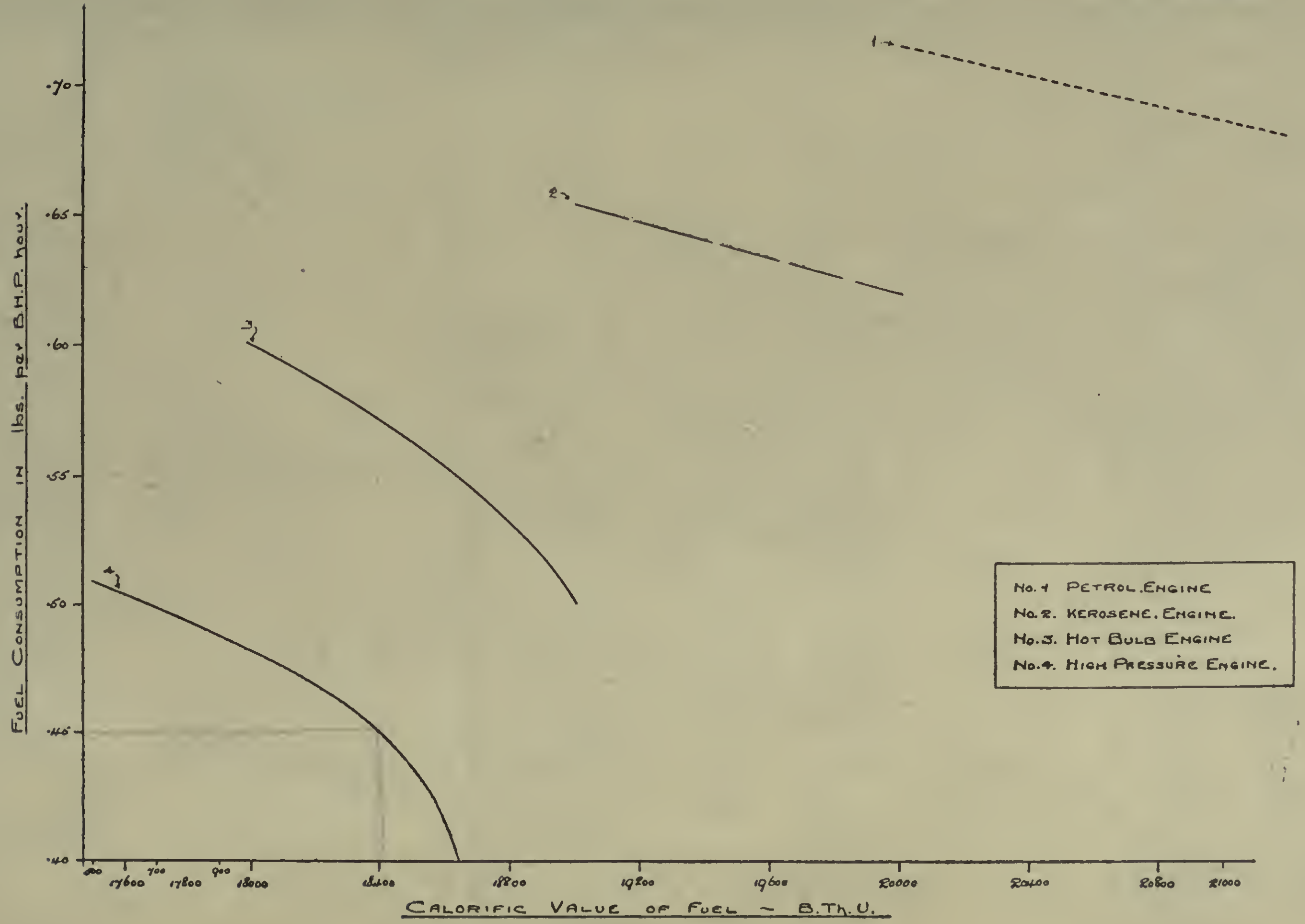
where H_2O is as before the water contents, and
K = a value depending on the specific gravity of the fuel.

The relationship between the specific gravity and K is given in Curve No. 4.


CORRECTION CONSTANT = K.
B.Th.U.
1,080
1,100
1,150
1,200
1,250
1,300
1,350
SPECIFIC GRAVITY.
.80
.82
.84
.86
.88
.90
.92
.94
.96
.98

CURVE NO. 4.

Curve No. 5.


Fuel Consumption in lbs. per B.H.P. hour.
·40
·45
·50
·55
·60
·65
·70
17600
700
17800
900
18000
18400
18800
19200
19600
20000
20400
20800
21000
Calorific Value of Fuel ~ B.Th.U.
1
2
3
4
No. 1 Petrol Engine
No. 2. Kerosene. Engine.
No. 3. Hot Bulb Engine
No. 4. High Pressure Engine.

In addition to the above formulæ the calorific value may also be approximately calculated from the chemical analysis of the fuel and the following formula is quoted:—

$$\text{Calorific Value} = 146\ C + 620\ \left(H - \frac{O}{8}\right) + 45\ S$$

where C = Percentage of Carbon.
H = Percentage of Hydrogen.
O = Percentage of Oxygen.
S = Percentage of Sulphur.

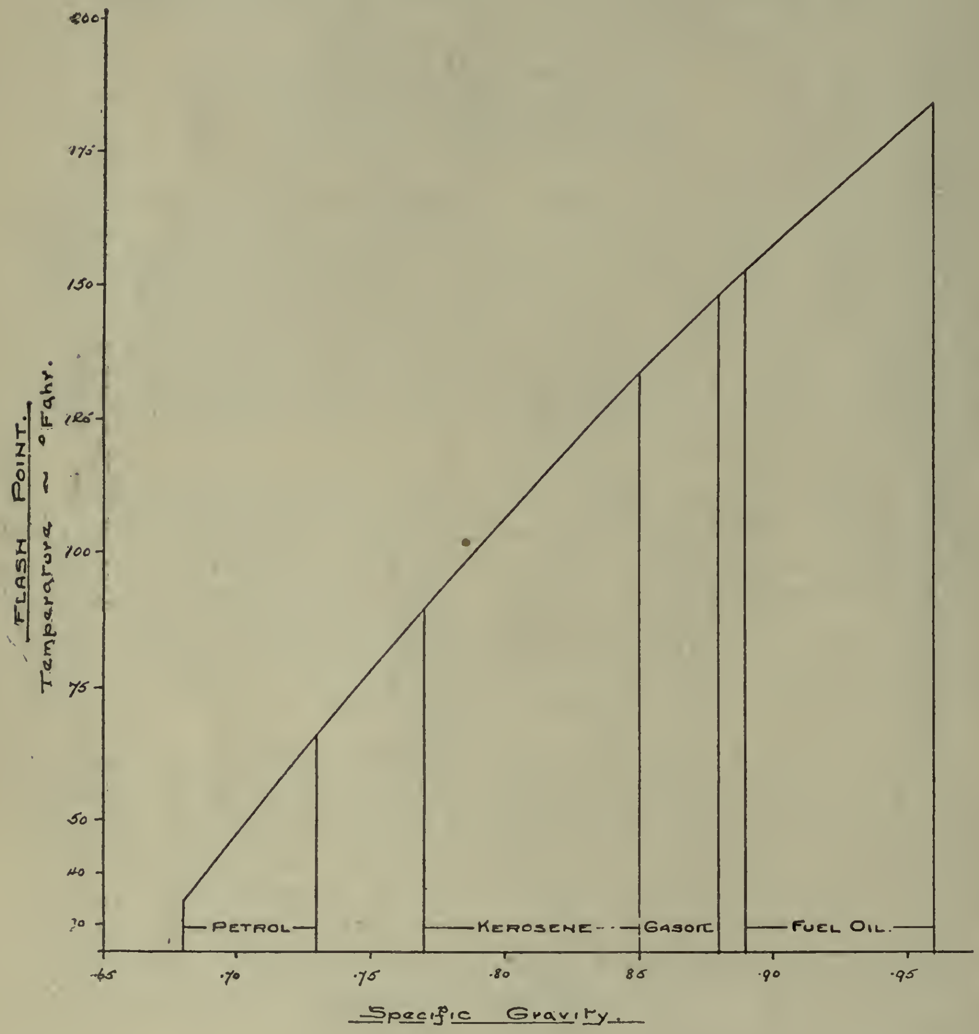
It is generally assumed that the fuel consumption for internal combustion engines is directly proportionate to the calorific value of the fuel and whereas such a statement is approximately correct for white oils and gas oils, actual tests have shown that in conjunction with fuel oils and residuums there is no direct relation. General data are given in Curve No. 5 for the various fuels and adjustments will have to be made depending on the type and size of engine cylinder.

3. *Flash Point.*—The flash point is of specific interest in connection with white oils, particularly so because white oils are usually employed in internal combustion engines with some type of external ignition device. For marine Diesel engines, where, as is well known, the fuel is ignited by the heat of compression, the flash point is of little value with regard to the determination of the suitability of the fuel.

Whilst from a technical aspect the flash point is no criterion as regards the suitability of a fuel, this value is useful to know in conjunction with the regulations which are laid down by the Board of Trade, Lloyd's and Local Port Authorities. It is stipulated fairly universally that a fuel oil must have a flash point above 150°F. In such cases where this regulation is not complied with, definite rules are laid down for the construction and fitting up of bunkering and piping systems generally on board such ships which carry low flash oils.

In Curve No. 6 the relation between specific gravity and the flash point is illustrated. The curve indicates the general tendency and is subject to adjustments depending on the base of origin of the various fuels under examination.

Crude oils have a comparatively low flash point, and in those cases where the uses of the volatile products are too small to make the refining of the crude oils a commercial proposition, the heavy marine oil engine offers advantages over the oil-fired boiler installation, inasmuch as the crude oil may be used with safety in an oil engine, whereas for an oil-fired furnace it is decidedly dangerous.



CURVE No. 6.

4. *Viscosity.*—The viscosity of a fuel can be expressed in Redwood, Saybolt or Engler terms, and the following approximate relation between the different values are given :—

Engler 1° to 6° = Redwood No. 1 ÷ 26·7 seconds to 31·3 seconds.

Engler above 6° = Redwood No. 1 ÷ 29·1 seconds.

Saybolt 1 second to 120 seconds = Redwood No. 1 ÷ 0·95 seconds.

Saybolt above 120 seconds = Redwood No. 1 ÷ 0·85 seconds.

In connection with white oils, viscosity has no importance as all fuels are sufficiently mobile whatever the grade or quality may be. In connection with black oils, however, the viscosity plays a very important part, inasmuch as heating may become necessary with very viscous fuels which in every other respect are quite suitable for use in engines. With these oils the viscosity also varies considerably with the temperature, and whilst any black oil becomes sufficiently mobile once it has entered the piping system on the engine, its viscosity under normal atmospheric conditions may be too high to enable the oil to be handled without any special preparations. As a result the viscosity of a black oil should be known in order to determine whether the fuel can be handled without heating or not.

Curve No. 7 shows the effect of temperature on the viscosity for a black oil which can be handled under all temperature conditions and another one which requires heating in order that it may be handled easily under similar conditions. For comparison a third curve is illustrated, showing the viscosity of a white oil.

5. *Fractionation.*—Fractionation or fractional distillation indicates the boiling point range and this value is useful for fuels which are vapourised prior to entering the combustion engine. This physical property applies, therefore, chiefly to white oils, and it is desirable that the boiling point range should be a minimum.

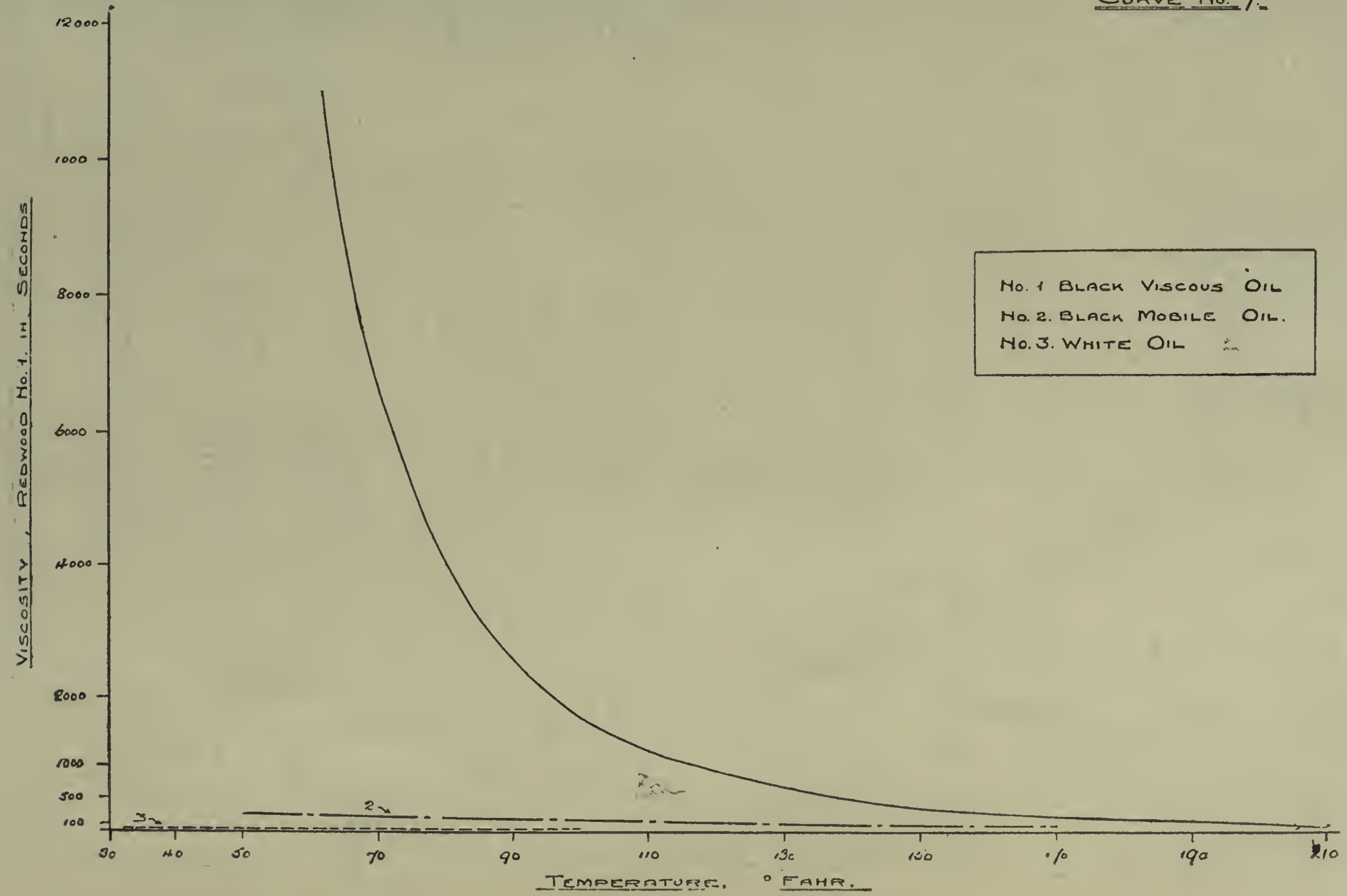
The following approximate data are quoted for the various grades of fuel:—

Grade of Fuel.	Boiling Point Range. Initial Boiling Point.		Boiling Point Range. Final Boiling Point.
Petrol	40°	to	200°C.
Kerosene	150°	to	300°C.
Fuel Oils	200°	to	400°C.

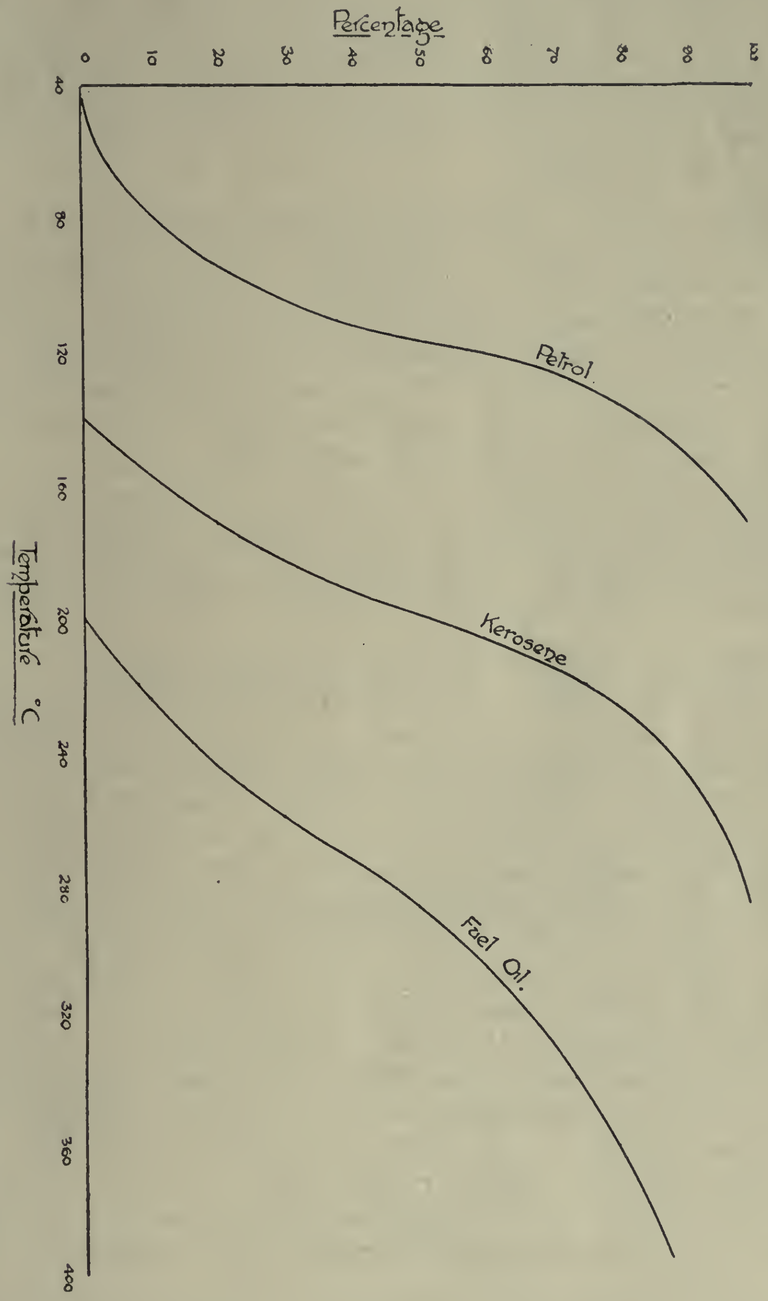
Curve No. 8 gives the distillation curves for various fuels indicating the general characteristics.

6. *Vapour Tension.*—The vapour tension is also a property which is of interest in conjunction with white oils as it is a measure of the volatility of the fuel, and from that point of view it indicates the ease when starting from cold condition. There is a close relation, of course, between the boiling point and

CURVE No. 7.


VISCOSITY, REDWOOD No. 1. IN SECONDS
12000
1000
8000
6000
4000
2000
1000
500
100
30
40
50
70
90
110
130
150
170
190
210
TEMPERATURE, ° FAHR.
No. 1 BLACK VISCOUS OIL
No. 2. BLACK MOBILE OIL.
No. 3. WHITE OIL
2
3

vapour tension, inasmuch as the lower the vapour tension the higher the boiling point. For black oils this property is of no practical interest.



FRACTIONATION CURVES
FOR
PETROL, KEROSENE AND FUEL OIL

CURVE No. 8.

7. *Mechanical Impurities.* — Whilst in connection with white oils there is very little danger of mechanical impurities being carried over with the fuel, the black oils, particularly on account of their viscosity, retain a certain amount of dirt and incombustible matter which it is advisable to filter out prior to using the fuel in the engine. It is, of course, desirable to get the percentage of such impurities to an absolute minimum, and by suitable filtering arrangements it is possible to remove all dirt and grit which may be contained in the oil prior to using the fuel inside the engine.

8. *Water Content.*—For white oils the water is easily settled out owing to the fact that the difference in the specific gravities of white oils and water is appreciable. Whilst in the case of Kerosene, a small amount of water will not introduce any technical difficulties, water in the carburetter of a petrol engine has been the cause of stoppages. As petrol is handled in bulk when used on board a vessel, the water question does not enter as it does in the case where petrol is distributed by cans.

In the case of black Oils, however, it is possible to retain water in suspension, particularly in those fuels of high viscosity. Filtering of the fuel itself is not sufficient in this case unless the oil is at the same time heated.

11. *Chemical Properties.* — 9. *Sulphur Content.* — This chemical property is of no importance in connection with white oils as the percentage of sulphur in white oils is negligible. Amongst fuel oils, Mexican and Californian oils contain percentages of sulphur. In former days it was contended that a sulphur content in an oil fuel would make that fuel unsuitable for use in internal combustion engines. It is admitted that during the chemical dissociation the hydrogen in the fuel combines with the oxygen in the air to form steam. Under certain favourable temperature conditions when the steam is condensed to water, it will, as such, combine with the sulphur content and form sulphur dioxide or trioxide. In that form it is capable of attacking the various engine parts. The prevailing temperatures are, however, invariablv above the boiling point of water and hence no corrosive action becomes possible. In the internal combustion engine low temperatures exist onlv on very rare occasions, for instance on starting from cold, and the corrosive effect of a sulphurous oil is therefore considered negligible.

In connection with hot bulb engines, water injection has always been a drawback when using a fuel containing sulphur, but it is satisfactory to know that the modern hot bulb engine has dispensed with this unnecessary and undesirable auxiliary arrangement.

10. *Ash Content.*—The ash content is of particular interest in the case of black oils. The ash content of the fuel is determined by the incombustible mineral matter which is retained in the fuel. Whereas with distillates, such as Gas Oil, only a very small amount of incombustible matter is carried over, the percentage in the case of residuums and heavy fuel oil may be considerable, and the nature and condition of that substance, which on combustion forms the ash content of the fuel, is such as to make filtration a most difficult proposition. It is, therefore, advisable to keep the ash contents of the fuel as low as possible, particularly owing to the abrasive effect which this ash may have on the moving parts of the engine.

11. *Asphalt and Coke Content.*—The tendency for a fuel to coke makes it unsuitable for engines, as the formation of an excessive deposit will necessitate frequent cleaning of the combustion chamber and the parts immediately in contact with it.

The Asphalt content of a fuel should be kept below a permissible maximum percentage, although the general tendency nowadays is to burn fuels with Asphalt content up to 30%.

Asphalt is a hydro-carbon compound, and can be burnt completely, provided the required temperature conditions are available.

Oils of an asphaltic base represent to-day a very large percentage of the total supplies of fuel oils, and it is pleasing to note that a certain section of the oil engine manufacturers have recognised this fact. The result is that to-day heavy asphaltic oils, such as Mex Fuel Oil, which have proved an excellent fuel for oil-fired furnaces, can be burnt in certain types of heavy oil engines.

Of late the progress of the internal combustion engine has been very rapid, and an oil which to-day is considered as entirely unsuitable for most engines will probably give excellent results in the improved models of to-morrow.

* * *

SESSION 1920-21.

President: LORD WEIR OF EASTWOOD.

Testing of Oils for Internal Combustion Engines

BY MR. T. M. McKENZIE, A.I.C., F.C.S.

Read Tuesday, November 16, 1920.

In order to fully understand the specifications for oil for internal combustion engines just given you, I think it would be interesting if I were to describe to you briefly the methods adopted for testing these oils, and the instruments used in such tests. Time will only permit me to describe the tests which are of general importance. The testing of petroleum and oil generally may be divided into two sections, namely, physical and chemical. As users, you will find that the physical tests are of much more importance than the chemical.

Before discussing the question of testing oil, it will be necessary to say something regarding the sampling of oil. What is a sample? A sample is a quantity representing the bulk of the oil or, in other words, a reduced scale of the contents of a tank, barrel or wagon. Unless we are sure that the sample is really representative of the bulk, it is a waste of time and money to carry out any tests, as such tests would really be the tests of that particular sample; whereas if the sample is a representative one, the results of the tests are the tests of the bulk of the oil. This is an extremely important point to remember. Sampling is one of the most difficult operations in connection with the testing of oil, yet, in the majority of cases, it is done by anybody handy—usually a workman. These remarks do not apply to oil only, but to any other material. For example, I think one of the most difficult things to sample is coal. For a number of years I was employed at a works where they burned, roughly, three to four thousand tons of coal weekly, so naturally their contract for coal was one worth having. In the course of my employment, I analysed several hundred samples of coal sent by coal owners and merchants; I could count on my ten fingers the times when the suppliers' samples agreed with the analysis of the bulk cargo. They were not, as you might imagine, always better—in many cases they were worse. They were really not representative samples. I wish to emphasise this point because it is a general belief that taking a sample is a simple matter, and I want to assure you that it is—as I have already said—one of the most difficult things to do properly. Sampling should always be done by trained men only, and for this reason—primarily the sample may be taken for some very simple reason, but one can never tell what the results of that sample may ultimately lead to, and if that particular sample is wrong the results may have very serious consequences. Having now explained how important it is to have the sample properly taken, I shall try to describe to you the correct way to do this.

It matters not whether the bulk be a barrel, tank, or even a steamer. The principle is the same in each case. If oil were really a homogeneous mixture, there would be very little difficulty, but this is seldom the case. For one thing, there is nearly always water present, and as water does not mix with oil under ordinary circumstances, and as the specific gravity of water is usually different from that of oil, we may find this water in pockets, or settled at the foot of the tank, or floating on the top of the oil, if the oil happens to be heavier than water, as with coal tar oils. To take a sample of this oil without the water or with too much water, would not be representative. The sample must contain the proportion of water that the bulk contains. One way to take the sample, if the bulk is not too deep, is to get a tube $\frac{3}{4}$ to 1 inch in diameter, fitted at the bottom with a wooden plug, from which a wire leads to the top of the tube. This tube is pushed slowly down to the bottom of the tank, the plug is pulled into place by the wire and fixed with a sharp blow on the bottom of the tank. In this way a " core " of oil is obtained. This is done from two or three parts of a tank or wagon, and all the " cores " mixed together thoroughly. If there are a number of such " cores " taken, the sample may be too large to take away, but this is easily overcome by thoroughly mixing the whole and taking a part as the sample. This method cannot always be adopted; for example, if the tank is 20 ft. high, this means you must have a tube at least 20 ft. long, and you can understand this would not be easy to handle. Very often the depth of a tank is over 20ft. In cases of this sort, one way is to have a loaded can fitted with a cork to which a wire is attached. The can is put into the tank to any desired depth, the cork pulled out, and then the can slowly drawn up and contents emptied into a receptacle. This is done at various depths, from the top to the bottom, and at different parts of the tank. The whole is thoroughly mixed and a quantity taken as the sample. This method, if carefully done, is quite efficient. Another method, which is only practicable when the oil is being pumped from one tank to another, say for example from the ship's tank to a shore tank, is to draw a small quantity periodically from the pumps. If this is done very frequently during pumping, a good representative sample is obtained. These are the methods generally adopted. Having obtained our sample, we will now proceed to test it.

As I have already stated, testing is divided into two sections, physical and chemical. We will deal first with the physical tests, and I shall endeavour to explain these tests in as simple a manner as possible.

Physical Tests.—To the layman, physical—and more especially chemical—tests are considered somewhat mysterious processes. Chemists are more or less looked upon as the alchemists were in the Middle Ages—men who carry out weird operations more or less—especially more—to the danger of the inhabitants generally. This is quite an erroneous idea. The operations we carry out are really extremely simple. All that is required is care and attention to detail, and of course practice. They look difficult and perplexing, but they are quite the reverse. Physical tests generally adopted for oil are:—

1. Specific gravity.
2. Flash point.
3. Viscosity.
4. Setting point.
5. Colour
6. Smell.
7. Boiling point.
8. Calorific value.
9. Water.
10. Ash.

In some cases distillation test.

1. *Specific Gravity.*—The specific gravity is the weight of a given volume of oil compared to the weight of a given volume of water at a given temperature. In this country we usually take this at 60°F. We determine specific gravity in two ways, depending to a certain extent on whether we wish our results to be extremely accurate or only approximately so:—

(*a*) *Hydrometer.*—This is simply a weighted tube fitted at its upper end with a graduated scale. Many kinds of hydrometers are in use, the simplest being that with which the specific gravity is read directly on the scale. (See Figure 1). The principle of the hydrometer is that the denser the liquid the more does a floating body project from its surface. Hydrometers are made in series, say from ·750 to ·800, ·800 to ·850 and so on. Hydrometers cannot be used in very viscous oils.

(*b*) *Specific Gravity Bottle.*—The most accurate method is by direct weighing of equal volumes of oil and water. This is usually done by means of a specific gravity bottle. This bottle is made of very thin glass, fitted with an accurately ground stopper. The neck of the bottle is narrowed below the stopper and has a

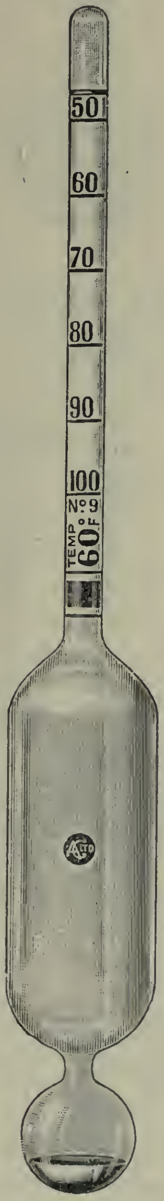


Fig. 1.

mark etched upon it to enable the operator to fill the bottle with the same amount of liquid each time. (See Figure 2). The volume used is usually 25, 50 or 100 ccs. The operation is carried out as follows:—The bottle is filled with water at 60°F. and carefully weighed. The bottle is now emptied, thoroughly dried, and is then filled with the oil to be tested, the temperature of the oil to be at 60°F. and weighed. The weight of oil, divided by the weight of the same volume of water, gives the specific gravity.

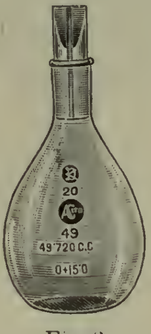


Fig. 2.

2. *Flash Point.*—The temperature at which an oil, on being slowly heated, begins to evolve a vapour in such quantity as, on the application of a flame, a momentary flash due to the ignition of the vapour occurs, is termed the flash point; and the temperature at which, on being further heated, the oil takes fire on the approach of a flame and continues to burn, is termed the fire point. There are a number of instruments used to determine this point, and as they are all similar in principle a general description of one will be quite sufficient to give an idea of the apparatus. (See Figure 3). This consists of a metal vessel or oil cup holding a fixed quantity of oil, which is placed in a hot-air chamber. The oil cup has a closely fitting lid or cover. This cover is perforated with several openings, which are opened or closed by means of similar openings on a sliding cover. The sliding cover is rotated periodically, and this uncovers the holes in the cover and at the same time causes a small test flame to project on to the surface of the oil. This is done at each degree rise of temperature until the flash occurs, and the temperature is then noted. Some apparatus are provided with stirring arrangements to stir the oil during the test.

Open and closed test.—In the former the oil is uncovered, and in the latter covered during the experiment. The temperature is considerably higher in the open test than in the closed test.

3. *Viscosity.*—The viscosity of oil is a measure of its fluidity at certain temperatures. This is an extremely important test for lubricating oils, and also for fuel oils. It is not so important for burning oils. There are many instruments used for the determination of viscosity, and, as in the case of the flash point

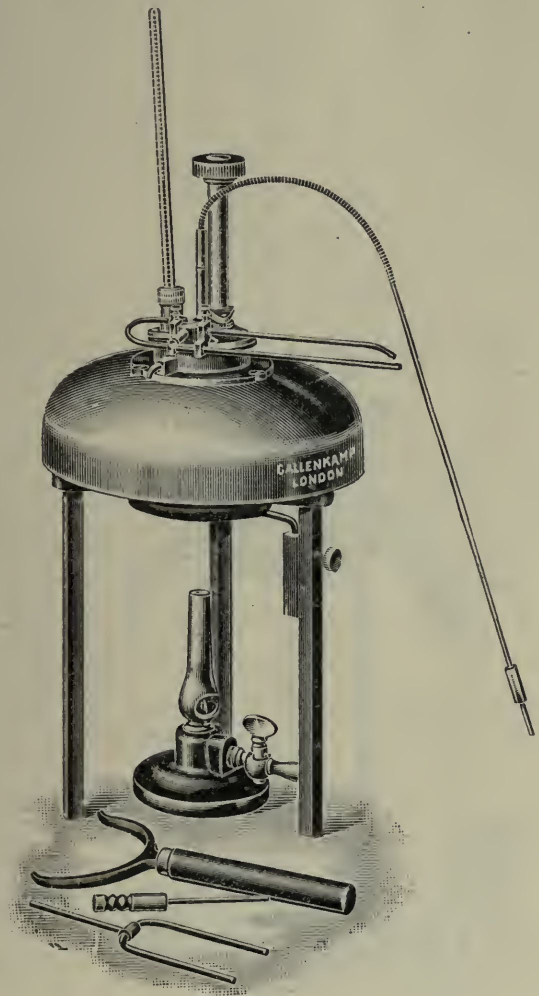


Fig. 3.

apparatus the principle of them all is, generally speaking, the same. In this country we use the Redwood Viscometer. (See Figure 4). Engler is used on the Continent, and the Saybolt in the United States. The figures given by all these instruments are arbitrary, and to a certain extent only can they be converted from one instrument to another. For this reason it is always necessary when reporting the viscosity of an oil to state the instrument used. As the Redwood is almost universally used in this country I shall briefly describe this instrument. It consists of a silvered copper oil cylinder $1\frac{7}{8}$in. in diameter and $3\frac{1}{2}$in. deep, having an agate jet fitted in a slightly conical metal seating. The cylinder is fixed in a copper water bath provided with a copper heating tube projecting downwards at an angle of 45° from the side near the bottom, as well as with a revolving agitator carrying a covered shield to prevent splashing, and a thermometer to indicate the temperature of the liquid in the bath. Water or oil may be used in this bath, depending on the temperature at which the viscosity is to be taken. The oil cylinder has a stopper, which is a small brass sphere which rests in a cavity in the agate jet. A thermometer

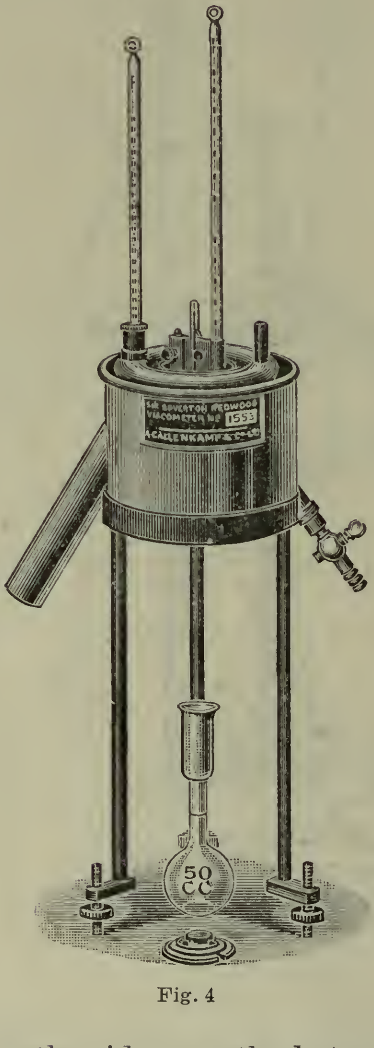


Fig. 4

is fixed in the oil cylinder. Inside the cylinder, at a short distance from the top, is fixed a small bracket, terminating in an upturned point which acts as a gauge of the height to which the cylinder is filled. Great care must be taken in the construction of the agate jet to secure uniformity. The test is carried out as follows:—The bath is filled with a suitable liquid and heated to the required temperature. The oil to be tested, which is poured into the oil chamber until its level just reaches the point on the gauge, is also brought to this temperature. A narrow-necked flask holding 50 ccs. is placed beneath the jet in a vessel containing a mixture at the same temperature as the oil. The stopper is raised and a stop-watch started at the same time. The number of seconds taken by the outflow of 50 ccs. is noted. This gives the viscosity in seconds on this particular instrument at the temperature recorded.

4. *Setting Point.*—This is the temperature at which an oil solidifies. This test is important. For example, it would be quite useless to use a lubricating oil having a setting point above 32°F. in a refrigerator or on machinery exposed to low temperatures. The oil to be tested is placed in a test tube. This is stoppered by a cork through which a thermometer is inserted. The tube is then placed in a larger vessel and the whole placed in a freezing mixture. The oil is examined at each degree fall in temperature, and that temperature at which it ceases to flow is the setting point.

5. *Colour.*—The colour of white oils is determined in this country by Wilson's chronometer (see Figure 5) which consists

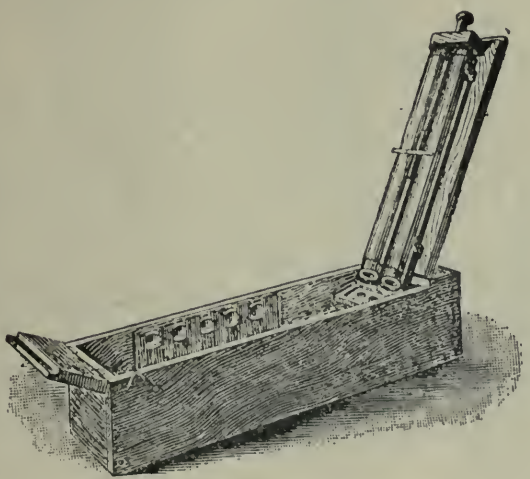


Fig. 5.

of two similar tubes 16 inches in length and closed at each end by a screw cap carrying a stout glass disc. Light is reflected upwards through the tubes by a mirror. One of the tubes is filled with the oil to be tested, and beneath the other, which is empty, is placed a disc of stained glass. On looking through the tubes the colour of the oil can be matched with the standard glasses. There are four standards: Water white, superfine white, prime white and standard white. The colour of lubricating oils is generally tested in a Lovibond's Tintometer.

6. *Smell.*—This test only requires practice.

7. *Fractionation.*—The fractional distillation is very important when testing white oils for the internal combustion engine. For example, a high boiling-point motor spirit would give difficulty in starting an engine. Again, a low boiling point motor spirit means a loss of a large amount of spirit by evaporation. To carry out this test the oil is placed in a clean dried distillation flask. A distillation flask is a round bottomed flask with a long neck from which projects a side tube. An example of this style of flask is shown on the lecture table tonight. The apparatus which is put together here to demonstrate the distillation of crude is similar to that which we use to carry out a fractional distillation of a motor spirit or kerosene. The distillation flask has definite dimensions, and in fact the various pieces of apparatus required for this test are all of definite dimensions, and any deviation from these dimensions causes a variation in the results. The flask is fitted with a cork, through which a thermometer is placed. The thermometer is arranged so that the bulb is just opposite the side-tube exit. A water-cooled condenser is attached to the side tube by means of a cork. At the end of this condenser is placed a graduated vessel to collect the liquid which distils over. Heat is now applied to the flask, and when the first drop falls from the side tube into the condenser the temperature is noted, and this is taken as the initial boiling point. Heating is continued until the flask is dry. The vapour condenses in the condenser and the liquid is collected in a graduated vessel. The temperature at which different quantities are collected is noted, until the distillation is finished.

8. *Calorific Value.*—This is determined in a bomb calorimeter. (See Figure 6). This apparatus consists of a steel chamber called the " bomb " which is enamelled in the interior to prevent corrosion. A small quantity of oil (usually about a gramme) is placed in a platinum capsule, which is placed in the

bomb. The bomb is fitted with a screw-down lid. The screw-down lid is furnished with a valve for admitting oxygen and another valve for the release of the gases formed by the combustion. The oil in the capsule is ignited by a current of electricity. This current is conducted to the oil by means of two

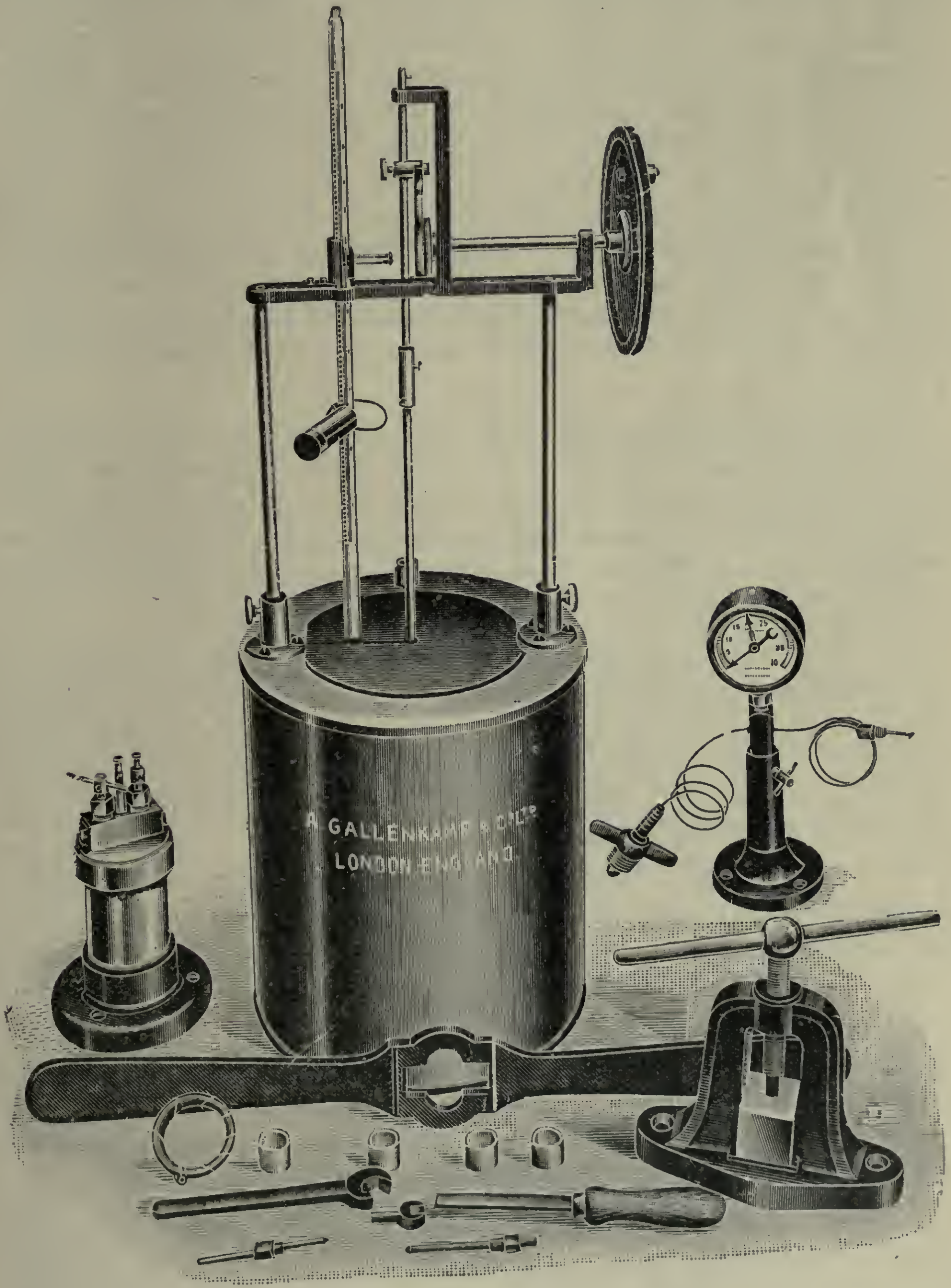


Fig. 6.

rods passing through the lid, which are thoroughly insulated from the lid. One of these rods carries the platinum capsule, and to complete the circuit, the end of each rod is joined by a thin iron or nichrome wire, which passes through the sample of oil. After weighing the sample, the platinum capsule is fixed to one rod and the lid screwed down into the bomb. Into the bomb is now admitted 20 atmospheres of oxygen. The bomb is placed in a vessel containing water. The temperature of this water is accurately measured and noted. This having been done, the electric current is passed through and causes the oil to ignite, thus raising the temperature inside the bomb. The heat developed is transmitted to the water outside the bomb, the rise in temperature is carefully measured and the calorific value of the oil ascertained from this data by a somewhat complicated calculation. In this test the thermometer is an extremely accurate and delicate one, capable of measuring 100th of a degree Fahrenheit.

9. *Water.*—The presence of water can be detected by means of an apparatus called a " Water-finder." This simply consists of strips of non-absorbent paper treated with a material which is soluble in water but insoluble in oil. These strips are put into the sample of oil and if water is present it dissolves the material on the paper, leaving the latter white.

To determine the quantity of water in a sample of oil is a somewhat different matter and there are two methods for doing this. One is to whirl a known quantity of oil in a centrifugal machine which causes the water to separate from the oil and the amount can be read off. Another method is to distil 250 ccs. of the oil and collect the water.

10. *Ash.*—The percentage of ash in oil is determined by igniting a sample of oil in a muffle furnace and weighing the ash obtained.

1. *Litharage Test.*—This test is applied to ascertain whether a spirit has been properly and sufficiently refined. To a small quantity of spirit litharge solution is added and the whole shaken for five minutes. If the spirit has been properly refined there will not be any alteration in its appearance; otherwise there will be a change in colour.

2. *Sulphur Test.*—The same apparatus and procedure is used as for the calorific value. The products of combustion are all retained in the bomb, and the sulphur is dissolved in a suitable

solvent and converted into a compound which is separated and weighed. From the weight of this compound the amount of sulphur in the oil is easily calculated.

3. *Bitumen Test.*—It is very difficult to accurately determine the percentage of bitumen in asphaltic oils. This is often done by heating a known quantity of oil to a certain temperature, but this method leaves much to be desired, as it is very difficult—if not impossible—to get concordant results by such method. Even when the following precautions have been taken, viz., to always use the same amount of oil, to heat it in the same vessel or a vessel of similar dimensions, to heat at the same temperature and for the same length of time, I have found the results obtained are very varied. The temperature necessary is so great that " cracking "—that is decomposition of the oil—always takes place, with the result that coke is formed. This coke is taken as being bitumen or asphalt.

A much more accurate method for determining bitumen is a chemical one. This is usually done by precipitating the asphalt from the oil with the use of Alcohol and Ether. A known quantity of oil—usually 2 grammes—is shaken with about ten times its volume of Alcohol and Ether mixture, 1-1, and allowed to stand for 24 hours at normal temperatures. The asphalt separates out and is filtered off. This residue is thoroughly washed with Alcohol and Ether mixture until free from oil, dried and dissolved in Benzene. The Benzene is then evaporated off, leaving the asphalt, which is weighed. This gives the amount of asphalt in the oil. With this test it is also necessary to take very special precautions in order to get concordant results, but if these precautions are taken, one is able to get results more in keeping with the actual percentage in the oil than by the other method

I think it would interest the audence if, in conclusion, I showed them generally the distillation of a crude oil and explained how various products are separated for further treatment. The apparatus which I am using is, of course, for laboratory work, and is somewhat different to that used in large-scale operations, but the principles involved are the same in both cases.

In this experiment the determining factor of the various products will be the temperature, but in large scale operations the determining factor is usually the specific gravity. The reason why I am using the temperature instead of the specific gravity is that, in order to determine the product by specific gravity one would require a very large amount, and that of course, as you will understand, is quite impossible in a laboratory experiment. Needless to say, time will not permit me to treat the various fractions sufficiently to show you the finished product, but you will see from this experiment enough to give you some idea of what is done on a works scale.

I should like to record my thanks to Messrs. Gallenkamp for so kindly allowing me to use their blocks as illustrations of this paper.

DISCUSSION.

A Member: There is one particular point upon which great stress has been laid: the calorific value. I should like to ask what is the relation of the calorific value of fuel oil, and the power, or explosion, of impulse conditions of that oil. The Calorific value has only been alluded to by the author as high or low, the high, involving the least quantity used per h.p. or s.h.p., and the low calorific value, the greatest quantity of fuel to do the same amount of work. The problem is that we get heat developed and an explosion inside a closed cylinder to push a piston up, or down, or in whatever the direction is desired; and then it has been found from experience that we must be careful to have cooling water round the cylinder in which the explosion is created. I should like a full explanation; also that the Author of the second paper would show us a pump attached to the calorimeter and state what allowance for difference of room temperature is made in that experiment.

A Member: We have been told of the flash point as having a variation of 9·5° by testers. Is there not a parallel here between the operator who can make clear ice and the one who makes cloudy ice? The experimenters on the flash point according to their carefulness and expertness in taking the temperatures, vary in degrees. On board a ship I cannot quite see how one can arrange, especially where there is any sea on, for demonstrating the points raised. In regard to some spirits, when being conveyed through the Red Sea, with the rise of temperature on the ship's skin tending to give a greater volume, is there any possibility of the temperature so rising as to alter the bulk of the spirit, to become a source of danger to the crew.

The future progress of the marine oil engine will largely depend on the energy and resourcefulness that the marine engineer will bring to bear upon this important problem.

We say to-day that the internal combustion engine will return as mechanical energy some 33 per cent. of the total available energy of the fuel. This is somewhat low, but so far thermodynamics have proved that it is never possible to recover even theoretically more than about 63 per cent. of the total available energy. So, if therefore you can convert 100 per cent. of heat energy into a possible 63 per cent. of mechanical energy, and if out of that we get a continuous return of 33 per cent. it means that we are getting a possible 50 per cent. of the available energy.

An average balance sheet for an internal combustion engine of the Diesel type would give the following data, allowing for the heat in the fuel as 100:—

	Per Cent.
Heat converted into Mechanical Energy	30
Heat in Exhaust Gases	25
Heat in Circulating Water	30
Friction	12
Radiation and Windage	3
	100

Whilst a considerable amount of study has been given to the chemical properties of oils I feel that the physical properties have been neglected, and it occurred to me that a further study of these physical properties will bring forth such information which will be extremely valuable both from the design and running point of view of internal combustion engines. The energy in the fuel is heat which has to be converted into mechanical energy, thereby passing through a good many phases which refer particularly to the physical rather than to the chemical properties. For instance, on a globule of oil being vapourised we are not quite sure as to what percentage of this vapour will be retained as permanent gas and what portion of it may be condensed into liquid again under certain unfavourable conditions. Furthermore, the ignition point of fuels under practical conditions is entirely different from the temperatures obtained with the present apparatus, and further information is necessary to obtain a clear picture of these properties.

I sincerely hope that the sea-going engineer will obtain such information and derive such benefits from this course of lectures as will enable him to become familiar with the different subjects relative to the design, construction and running of the internal combustion engine.

The Author: A Member has asked about the room temperature when using the bomb. We take the temperature of the room and record it. The water in the outer vessel is at the room temperature and the difference between that temperature and the temperature generated by the explosion inside the bomb gives us the figures from which we calculate the calorific value. With regard to the flash point, I am not sure that it is the human element that makes the difference. I think it is more probably due to the fact that we have not yet got the perfect apparatus. The tests obtained are usually pretty much the same when using the same apparatus. If we were able to evolve an apparatus in which the perfect conditions could be obtained, then the different operators would get concordant results. Another reason why I do not think it is due to the operators is that, with skilled operators naturally great care is taken in carrying out the tests, and yet we get these differences, and the differences are not always in the same direction. With regard to the expansion of oil, say in the Red Sea, this is allowed for in the expansion trunks. The coefficient of expansion of the oil is known, and allowance is made for this, so there is no danger to the ship or crew.

Notes.

The following note was received from one of our members, bearing on the subject of oils:—

INVINCIBLE OIL ENGINE (TWO-STROKE HOT BULB).

Particulars of Test 25 B.H.P. Single Cylinder Engine on Palm Oil.

The oil was heated to a temperature of 160° F. and at this temperature had a specific gravity of ·895.

The oil flowed at a temperature of 90°F. and flowed freely at 100°F. After a no load test, the engine was put under load to run for a continuous test of six hours under full load, the horsepower developed during its trial was 22 B.H.P., a drop of 12½ below the power developed on fuel oil.

The consumption was ·63 of a lb. per B.H.P. per hour, this is slightly higher than fuel oil, the latter being ·56 of a lb. per B.H.P. per hour.

After the trial, the hot bulb and cylinder head were examined, and it was found that instead of a black carbon deposit, there was a very slight deposit of brown powder, this deposit was not at all gritty and was not of such a nature to harm the engine.

SESSION 1919-20.

President: LORD WEIR OF EASTWOOD.

The Development of the Internal Combustion Engine.

BY MR. CHAS. BAXTER.

Read Tuesday, October 7, 1919.

IN marine engineering the steam reciprocating engine has in the past held undisputed sway, and along with the steam turbine will undoubtedly hold a prominent position in the future.

Signs are not lacking, however, that at the present time the internal combustion engine is receiving considerable attention in marine circles, and what is more, shows every indication of becoming a serious rival to the two previously mentioned prime movers, it being at last looked upon by engineers as a serious proposition.

In the early days of the development of this new prime mover, wild statements were made by its over energetic sponsors in order to cover up the many faults it then possessed.

It was stated that the reliability was at least equal to the reciprocating steam engine and on most other points with perhaps the exception of the starting difficulty it was said to outrival its opponent. There is therefore no getting away from the fact that these irresponsible statements did an infinite amount of harm, for when the purchaser found that they could not be substantiated, the pendulum swung the other way and a period of unnecessary scepticism prevailed which took many years to dispel.

Early engines of this type were very unsatisfactory in many ways, all of which, directly or indirectly, affected their reliability and due to the small sizes in which they were then built, no attempt was made to boom them from a marine point of view, which is perhaps very fortunate for the marine engineer as the development through which all prime movers have to pass to reach the commercial stage, then took place on shore which was in this case a far safer proceeding.

THE DEVELOPMENT OF THE INTERNAL COMBUSTION ENGINE.

Problem after problem was solved, however, by close application, which incidentally enabled larger sizes to be built, until now we find successful ships of reasonable size, engined with this prime mover running with a regularity which was only previously associated with steam.

These ships, although not very large in either size or number, are rapidly growing in both directions and it is undoubtedly a sign of the times, which augurs well for their success, that marine engineers are beginning to be interested.

The development taking place upon shore put marine engineers rather at a disadvantage, as it was only at intervals that many of them got the opportunity of coming into contact with these engines. They were therefore left rather in the dark as to the historical and development side of the study, and when this branch of engineering is taken up it is found that it has already grown to almost gigantic proportions; engines differing widely from each other being used for all manner of purposes, ranging from the propulsion of ships to the propulsion of aeroplanes and motor cycles.

Now although the members of the Institute are chiefly interested in the propulsion of ships it is a fact that the development of this engine in its different branches all went to form the whole, it being impossible to say where the discoveries and developments in one branch cease to influence another, the illustrations appended indicating the diversity in design.

It will therefore be found upon taking up the study of this prime mover that the particular knowledge required is scattered over innumerable text books and technical journals, the perusal of which is truly a formidable task, and it is with a view to helping engineers interested in the subject that the author has been tempted to offer this paper.

The information given has been carefully compiled during the past few years, and if it saves some of the time usually spent in wading through ancient and modern literature upon the subject, this effort will be amply repaid.

The present paper is not the first one before the Institute upon the internal combustion engine, many valuable ones preceding it, which are well worth the time spent in perusal, and if the discussion upon the present one is as practical and enlightening as those on its predecessors, much valuable information will come to light, the views of members always meriting considerable respect.

THE DEVELOPMENT OF THE INTERNAL COMBUSTION ENGINE.

Historical.—The very early history or origin of this engine is rather obscure and seems to have begun with a period of speculation, before anything was done, the Abbé Hautefeuile described an engine in the year 1678; a partial vacuum caused by burning gunpowder in a cylinder, raising water, the resulting gases being cooled. Gunpowder in small quantities was exploded in a cylinder, the pressure being allowed to escape through non-return valves; upon the gases cooling and the pressure falling below that of the atmosphere the piston was forced down by atmospheric pressure.

According to patent records which are something of a guide in the matter, a Robert Street, of this country, in May, 1794, described an engine which worked by turpentine.

Other atmospheric type engines such as that by Samuel Brown, 1823, a beam engine, were made, in which the working depended upon the exploded charge being cooled and so forming a vacuum, allowing the atmosphere to do the work.

The first engine to attain any success however was that of Lenoir, 1860, a diagrammatic sketch of which is shown in Fig. 1 along with its diagram.

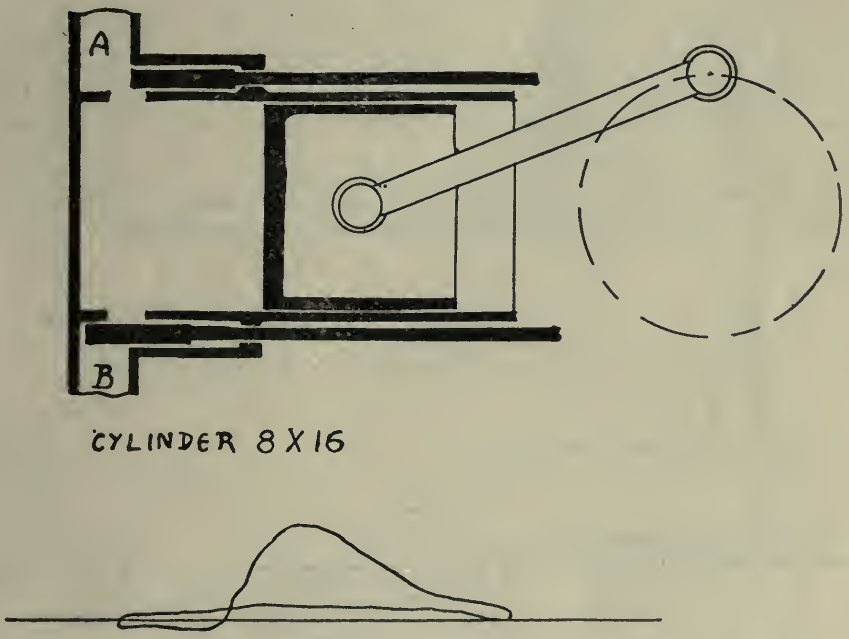


FIG 1

Cylinder 8 inch Bore by 16 inch Stroke.

This is shown as single acting, but in reality it was a double acting engine, very similar to the steam engines of that day.

A slide valve A, after the style of the usual D valve, uncovered the inlet port as the piston began to move outward and a charge of gas and air was inducted for about half the stroke, the valve then closing; the charge was ignited by means of an electric spark, the resulting pressure propelling the piston. At the end of the stroke the exhaust valve B opened and the piston on returning swept out the burnt charge. The impetus thus given to the fly-wheel allowed the engine to give off a certain amount of power and tided it over the idle stroke. The maximum pressure was in the region of 50lbs. per sq. inch and the consumption is said to have been 95 cubic feet per indicated horse-power hour.

An engine afterwards made by Hugan seems to have been of better mechanical construction, as the consumption was reduced to 85 cubic feet per indicated horse-power hour, whilst an atmospheric type engine by Otto and Langen further reduced the consumption to 44ft. by allowing the mixture to come in below a slightly raised piston and exploding it, the piston being rapidly sent upwards like a projectile, thus giving favourable conditions by rapid expansion, the weight of the returning piston along with the atmosphere, doing the work. The feature of Otto and Langen's engine was that the piston was free on its upward stroke and was connected to the crankshaft on its return stroke by a free wheel clutch. The maximum pressure was about 56lbs. per square inch, the consumption in large sizes falling to 30 cubic feet. Fig. 2 shows the diagram.

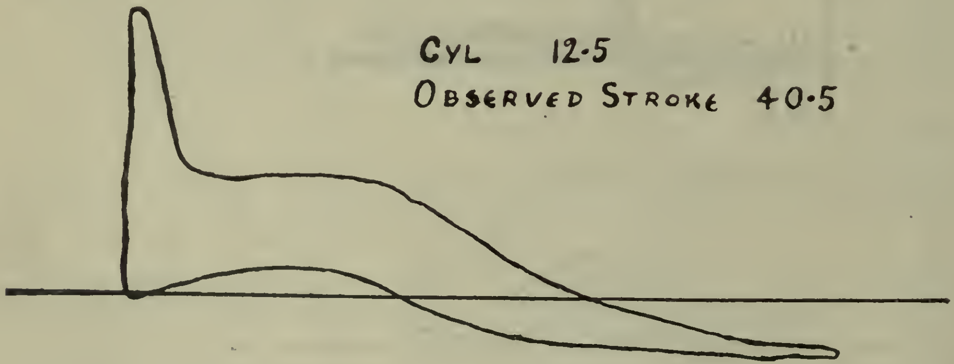


FIG 2

THE DEVELOPMENT OF THE INTERNAL COMBUSTION ENGINE.

It may be of interest to note that an engine by Samuel Brown was run experimentally in a boat on the Thames, this being so far as the author is aware, the first internal combustion engined boat.

The next step of any note in the chain of events was the discovery that if a charge of mixture be drawn into the cylinder, compressed by the piston, exploded, thus doing work upon the piston and driven out on the piston's return, that this formed a much more efficient cycle.

Four Stroke Cycle.—The cycle of events was thus completed in four strokes, from whence it derives its name, it being without doubt the point at which the real history of the internal combustion engine began.

This type shown in Fig. 3 was patented by Beau de Roches in 1862 and made by Otto in 1876. Referring to point A on the diagram, the inlet valve opened and the piston proceeded upon its outward stroke drawing in a charge of mixture, the inlet

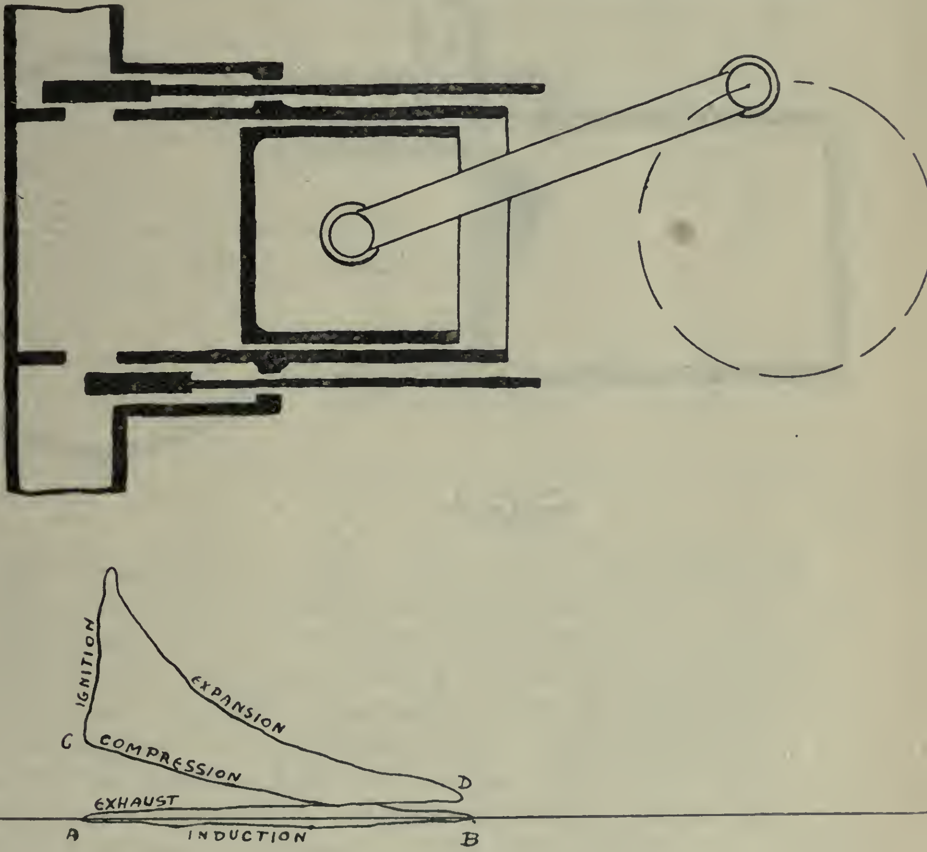


FIG 3

valve then closing at B and the piston returning compressed the mixture to C, where it was ignited on the inner dead centre. The rise in pressure due to the explosion propelled the piston on the power stroke, the exhaust valve opening at D, and the piston returning driving out the burnt gas, starting the cycle again at A.

At first sight it may seem strange that anything can be gained by compressing the mixture previous to ignition, but it is found in practice that there is a great gain in efficiency, the mixture in modern engines being compressed to as high a pressure as practicable, as will be explained later.

Two Stroke Cycle.—The two cycle engine also compresses its mixture previous to exploding it and is now in extensive use, the cycle of operations being thought out by Sir Dugald Clerk. The engine is shown in Fig. 4 and functions as follows, completing its cycle of course in two strokes.

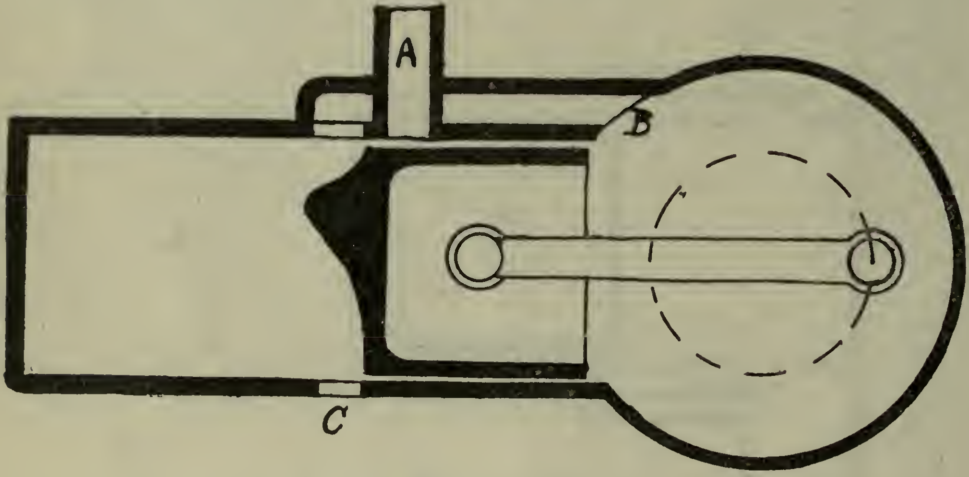


FIG 4

The piston on its in-stroke causes a vacuum in the crank chamber which fills with mixture when the piston uncovers the inlet port A. It also compresses the previous charge in the top of the cylinder which is ignited upon the inner dead centre, thus propelling the piston and compressing the charge beneath it, previously drawn into the crank case. Near the end of the stroke the exhaust port C is uncovered and the transfer port B, the new mixture driving out the products of combustion and filling the cylinder, the piston crown being shaped to facilitate this. It will be seen that the engine has its expansion, exhaust and inlet strokes all in one so to speak, the other stroke being for compressing the charge. These engines can be made without valves and are simple and reliable, being used in great numbers for small craft and motor cycles.

It should be borne in mind that they give more power than the four cycle, exploding twice as often, but not twice the power, as a proportion of the stroke is taken up in uncovering ports. These engines will also run in either direction, but as the very small ones are not self-starting, this is not of much consequence and a reversing gear is provided. They are also not as efficient as the four stroke, due to some of the inlet charge going out with the exhaust, although where the scavenging is done by air this does not of course apply. The uncommon feature about the engine is that the inlet charge drives out the exhaust, but this is found in practice to be quite satisfactory, the only precaution being the placing of a wire gauze in the transfer port to prevent accidental explosion in the crank chamber. Another point is that when running upon a partially closed throttle the cylinder is not quite filled with mixture and miss-fires, the next charge filling it and allowing the explosion to take place. It is possible for this to happen with great regularity, the explosion every other stroke being known as four stroking, but normal functioning is at once resumed when more throttle is given.

Four Cycle Diesel.—In this type a charge of air is drawn in on the induction stroke and compressed on the compression stroke to give it a temperature in the region of 1,000° F., which is higher than that required to burn the fuel.

At the end of the compression stroke the fuel valve opens and oil is injected by compressed air, thus burning automatically, the expansion and exhaust strokes taking place as in all four cycle engines. The fuel is injected by air in order to pulverise it and the extended burning period gives a diagram more like that of the steam engine, see Fig. 5.

All kinds of heavy fuel are used successfully in these engines and no ignition apparatus is required, but the heavy strains set up have been a formidable obstacle to designers.

Two Cycle Diesel.—This is identical with the two cycle engine previously described, but on the Diesel system. Assuming the piston to be at the end of the expansion stroke a charge of air is admitted under pressure, driving out the burnt gases and filling the cylinder, the piston then compresses the air to give the high temperature required, the fuel being admitted, burnt, expanded and then driven out by the next charge of air.

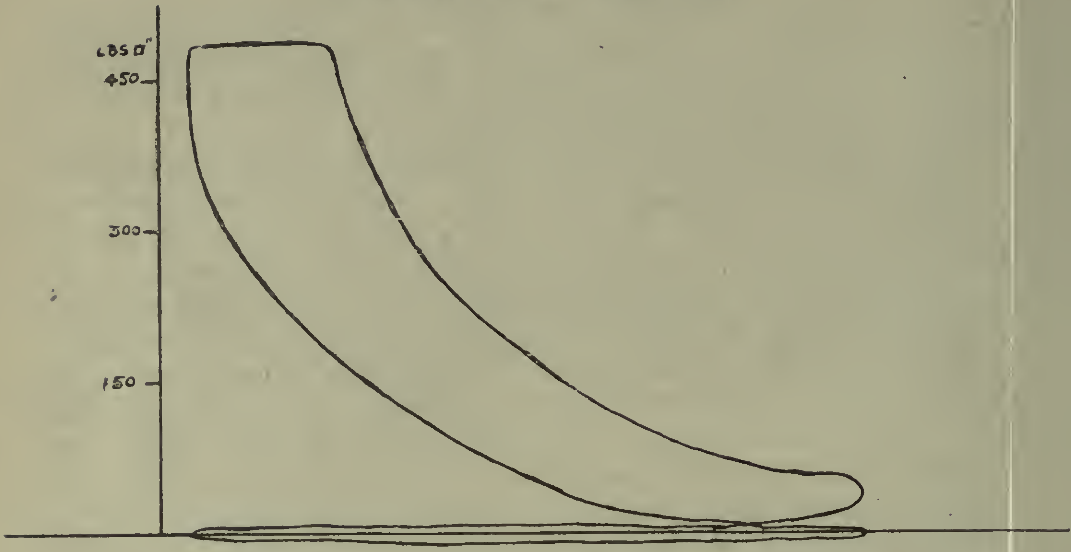


FIG 5

Semi Diesel.—This is an engine coming into extensive use, because it practically works upon the Diesel principle but avoids the higher compression pressure. It is also sometimes named by its makers the "Hot Bulb Engine," this term being considered by some to be more appropriate, the Diesel cycle being slightly modified.

In Fig. 6 it will be seen that the cylinder is water jacketed in the usual way with the exception of the hot bulb, which is thus allowed to attain considerable temperature. A charge of air is taken in on the induction stroke and compressed on the next stroke to quite a reasonable pressure, the ignition of the oil taking place by it being injected into the bulb, the heat of which augmented by the heat of compression is sufficient to burn the oil.

These engines may of course be constructed to operate upon either the two or four cycle principle, but by far the greater number operate upon the two cycle principle, the fuel being injected on the solid system, *i.e.*, by pump without compressed air.

Review.—It will be found that although the engines met with differ widely in detail and arrangement of parts, they may all be classed under one of the previously described headings. It is difficult to classify them according to their uses as certain types of engines are found in many spheres, but the following will give some idea of their arrangement in this respect.

Two Cycle Engines.—Motor cycles, small craft, large engines operating upon blast furnace gas, lighting sets, etc.

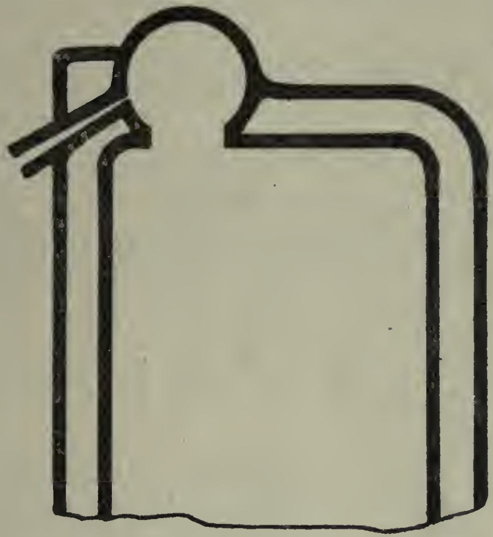


FIG 6

Four Cycle Engines.—Motor cycles, cars, stationary engines, aeroplanes and airships.

Two and Four Cycle Diesel Engines.—Marine and generating station work.

Semi Diesel Engines.—Small craft, etc., where simple operation is imperative and the speed is fairly constant.

Other Cycles.—From time to time other cycles make their appearance, such as the Six Cycle, where a charge of mixture is inducted, compressed, exploded, exhausted, then a charge of clean air inducted and then exhausted, the idea of course being to get rid of all products of combustion. These however do not seem to be put to any commercial use, but a cycle proposed by Mr. Durtnall, one of our members, is interesting and should be noted. Fig. 7 shows the diagram and it will be seen that air at atmospheric pressure is taken in from A to B, the inlet valve then closing and the pressure falling to C. The compression stroke then follows the line CBD, the pressure at the end of this stroke being sufficient to ignite and burn the fuel on injection from D to E, the expansion following the line EFG.

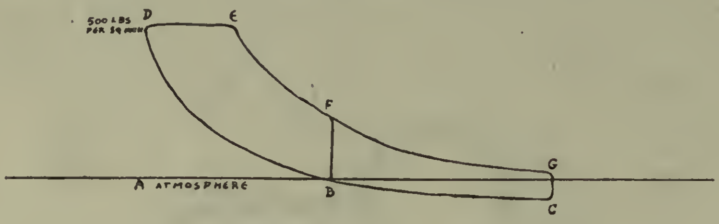


FIG 7

The vacuum at the end of the admission stroke gives a cushioning action and the expansion line falling to atmospheric pressure would apparently give a high theoretical thermal efficiency.

Compounding.—Attempts have been made at compounding internal combustion engines, but without much success, the power derived from the L.P. cylinder hardly balancing the extra frictional losses.

Double Acting Engines.—These are constructed by some makers in very large sizes for operating upon such fuels as blast furnace gas, the general lay-out greatly resembling steam practice, some types having their cylinders arranged in tandem.

Operation and Construction.—Cooling Pump.—Water jackets are fitted to cylinders in order to keep the temperature within practical lubrication limits, about 200° F., the circulation being promoted by means of a pump, often of the centrifugal type, or by means of a connection termed Thermo Syphon. In this case particular care has to be paid to pipe resistance, having the pipes of large diameter and all bends of easy radius. In large engines, valves, pistons, and rods are made hollow and water circulated through them.

Cooling Air.—Small automobile and other type engines are often air cooled, fins being cast on the cylinders to assist radiation.

Radiators.—Where only small quantities of water are available, radiators are provided to cool the water, often consisting of gilled tubes through which the water is circulated, a good air draught being produced by a suitable fan.

Cooling Hopkinson Method.—Professor Hopkinson of Cambridge carried out some research work upon cooling and found that he could quite successfully cool engines by spurting jets of fresh water upon the parts inside the cylinder which were likely to get too hot, such as the exhaust valve, thus keeping them at a reasonable temperature and allowing the engine to continuously function normally. This system is interesting as it differs widely from any current practice, but so far as the author's knowledge goes it has not yet been adapted commercially.

Lubrication.—No startling innovations are noticeable in this respect with the exception of the small high speed type, which has forced feed through hollow crankshaft and connecting rods, the pressure used often being high, about 40lbs. per sq. inch. Very small two cycle crankcase compression engines are lubricated by mixing the oil with the petrol in the fuel tank, but the medium and larger sizes are fitted with a more positive system such as by gear or plunger pump.

Starting.—The starting difficulty is a great drawback, which applies to all types of internal combustion engines, and all manner of means have been tried to remedy the defect. One idea used on stationary engines is to put the engine upon the firing stroke, pump in a charge of mixture by hand and then ignite it. Many of them like the one just described are tedious and rather uncertain and a method extensively used is by means of compressed air, a diagram of the arrangement being shown in Fig. 8.

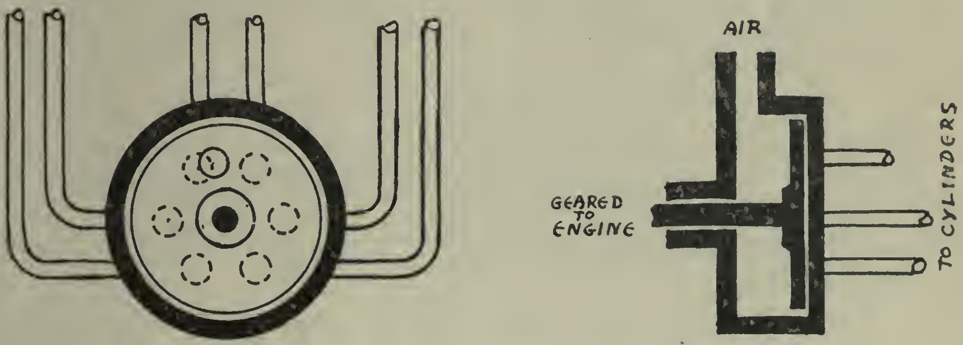


FIG 8

This consists of a rotating disc valve positively driven from the engine and timed to deliver air under pressure to each cylinder as it comes upon the firing stroke, the air working the cylinder as in steam practice. It should be noted that the air steams the cylinder on its firing stroke and is exhausted in the usual manner, the inlet and compression strokes not being interfered with. A non-return valve is fitted to the air pipe at each cylinder so that when the engine takes up the firing the valve closes and prevents the explosion pressure entering the air pipe, the air then being shut off by hand.

Easy starting in large engines is obtained by sliding a differently profiled cam into position which reduces the compression pressure by delaying the closing of the inlet valve until the piston has traversed part of the compression stroke, thus returning to the induction pipe a proportion of the mixture taken in.

Small automobile type engines are very often started by means of an electric motor supplied by current from an accumulator, many makers fitting this system as standard.

Reversing.—This is often accomplished by sliding the camshaft endwise, bringing another set of cams into operation and restarting in the opposite direction by air.

Many types of two cycle engines reverse by slowing down and then causing preignition by injecting the fuel too early, so causing what is commonly termed a "back fire," which stops and restarts the engine in the opposite direction.

Governing.—Various methods of controlling the speed of engines are in vogue: in the case of:—

Diesel type engines for instance, this is done by varying the oil injection period, although the minute variation of the exceedingly small quantity of oil injected per working stroke entails considerable care in design and accuracy of workmanship, in constructing the pumping apparatus.

Hit and Miss.—Stationary gas engines fitted with heavy fly wheels were for years governed by cutting out an explosion now and again, known as "hit and miss," governing the governor causing the gas valve to miss opening altogether when the speed became too high.

Quality.—Modern times demand a smaller speed fluctuation and so quality governing by varying the gas valve lift is in vogue, which, however, is likely to result in weak and slow burning mixtures with resulting high loss to the cylinder walls, thus unfavourably influencing the thermal efficiency, unless carefully arranged.

Quantity.—Governing by controlling the quantity of mixture inducted also has considerable vogue, especially upon engines used upon small craft, motor cars and aeroplanes; where it is universal, the quantity being varied by a throttle in the ordinary way.

Governing on the throttle is not good from a theoretical point of view, as it results in the cylinders being only partially filled with mixture, the engine thus running under inefficient conditions, but apart from this, it is very satisfactory and often most convenient.

Pressure Fluctuation.—In gas engine work an elastic bag is fitted on the gas line to provide a reservoir of gas and prevent fluctuations in gas pressure, acting as a sort of air vessel.

Indicating.—All engines may of course be indicated, an optical type of indicator being used for the high speed variety, but generally speaking this is not done with the same ease and certainty as in steam practice, some type of dynamometer brake being a more reliable method of testing the condition of the high speed type.

The "Clerk" indicator has now been considerably improved and the writer is informed by Sir Dugald Clerk that a description will be published shortly; this will certainly be looked forward to with great interest by engineers interested in the subject. The compression and explosion pressures may easily be obtained by means of the "Okill" indicator, Fig. 9, the thimble being screwed down until the movement of the valve can just be detected, the pressure then being read off on the scale.

Silencers.—A study of indicator diagrams will show that the exhaust valve opens some few degrees before dead centre and releases the gas at considerable pressure, which results in noise. An expansion chamber is therefore fitted on the exhaust line to minimise the sudden fluctuation in pressure, the art of designing this chamber consisting of arranging it so that the gases are gradually expanded without putting back pressure upon the engine.

Cylinders.—The cylinders on small engines are cast solid, the larger types being fitted with detachable heads and separate liners. Many aero engines have their cylinders turned from solid billets, or cast in aluminium with steel liners fitted, a 5in. cylinder for instance having the aluminium block raised to 340° C. and the liner dropped in, the latter having a finished size 0·005in. larger than the block.

Pistons.—Large types are fitted with the usual junk rings, the small types being solid, an extra ring being provided near the bottom of the skirt to act as a scraper ring and prevent oil passing the piston and carbonising in the cylinder head. Many makers pin their rings in position to prevent the slots working into line.

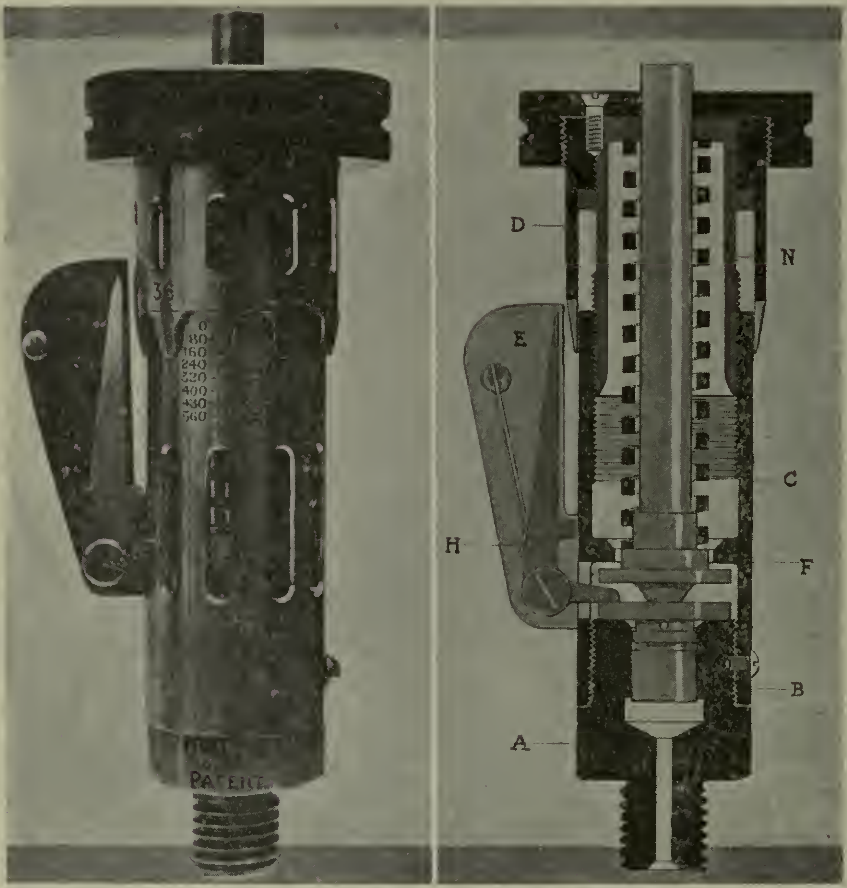
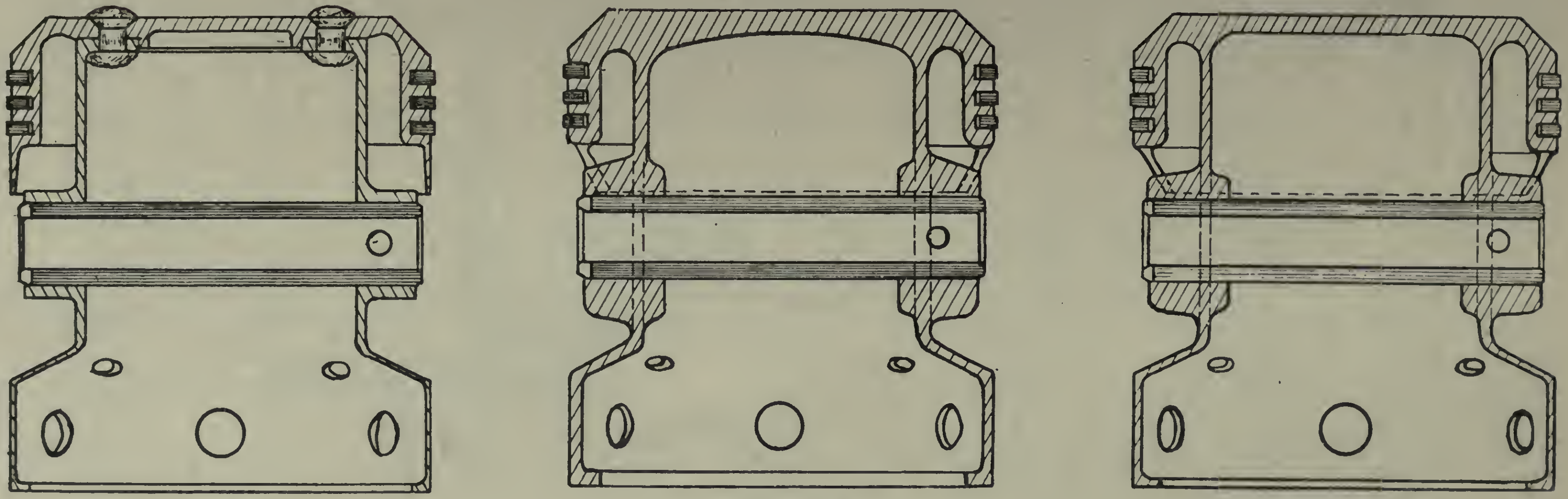


Fig. 9—The "Okill" Indicator.

In high speed type engines where the losses, due to inertia, become serious, special attention has been given to the construction of lighter pistons. it being found that a reduction of 10% in piston weight, for instance, influences the horse power developed considerably, being 12% on the higher reaches of the power curve, in an everyday example. The matter having been investigated separately by two engineers, E. Talbot and H. Ricardo, the two types shown in Fig. 10 and Fig. 11 have been



STEEL. ALUMINIUM. CAST IRON.

Fig. 10.

evolved, the Talbot piston Fig. 10 known as the "Zephyr," being by far the earliest, enjoying an immense popularity, the steel one being very light indeed.

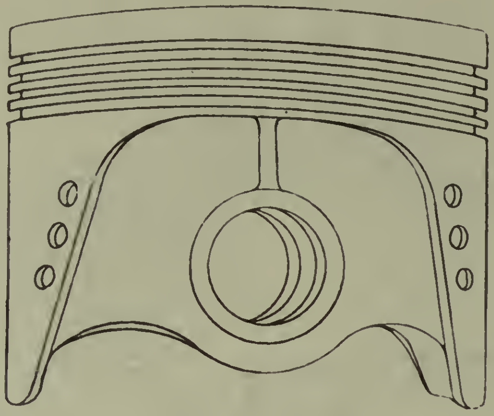


FIG 11

Aluminium pistons have been used with great success in many engines, the aluminium getting rid of the heat quickly, due to its higher conductivity, but greater clearance having to be given, due to the higher coefficient of expansion. The coefficient of expansion of iron being 556×10^{-8} and that of aluminium 1234×10^{-8} per degree Fahrenheit.

The majority of engines, being single acting, have their pistons functioning as a crosshead, this being quite satisfactory in practice, aero engines giving no trouble with a piston length over bore ratio as low as 0·6 with a maximum connecting rod angularity of 16°.

Gudgeon Pins and other important parts are now made of high grade alloy steels, the H section connecting rod being very popular.

In stationary gas and oil engine practice the same rules apply to connecting rod bolts as to chains, insurance companies stipulating that they should be annealed periodically.

Gearing.—As the valves on a four cycle engine are operated every two revolutions, a two to one gearing must be provided and this unfortunately often gives the engine a very complicated appearance, operating as it does a number of other mechanisms such as the ignition, fuel pumps and starting gear.

Neatness of design is now a telling point from a sales point of view, and such matters are receiving more attention than hitherto.

Carburation.—It will have been noted that an engine of the internal combustion engine type can be run upon any substance which will form an explosive mixture, history telling us that some of the earliest were run upon gunpowder.

Modern engines function satisfactorily upon all kinds of fuel, from heavy oil to acetylene gas, the proportion of air of course depending upon the value of the fuel.

In gas engine practice, a gas valve is provided in the air intake which lifts at the same time as the inlet valve, thus mixing the gas with the air. In oil engine practice the oil is often injected directly by a small plunger pump into some part of the combustion head which is allowed to acquire considerable heat, the oil being thus vapourised upon the induction stroke, the part being heated for starting purposes by means of a lamp.

For light oil and spirit engines where simplicity is the main point, the idea shown in Fig. 12 has been used, the fuel being drawn in along with the air, upon the inlet valve lifting.

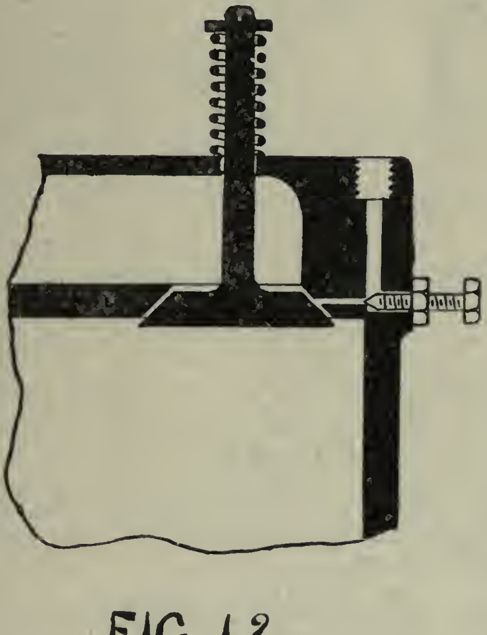


FIG 12

Small petrol and paraffin engines form their mixture in a carburetter, a diagram of which is shown in Fig. 13. A float chamber A keeps the level of the fuel just below the top of the jet B. Upon the induction stroke, air is drawn in at C and fuel out of the jet B, the mixing taking place in the pipe.

In the case of paraffin the inlet pipe is heated externally by means of the exhaust, in order to assist vapourisation.

This type of carburetter is only satisfactory for constant speeds as it can be set to give a correct mixture for any given speed by altering the jet, but it is found that for a variable speed engine such as is used in automobile work some design is needed which will give an approximately constant mixture at all speeds. The reason why the mixture varies with the speed is because the laws of flow of air and fuel are not the same, the mixture getting richer as the speed increases.

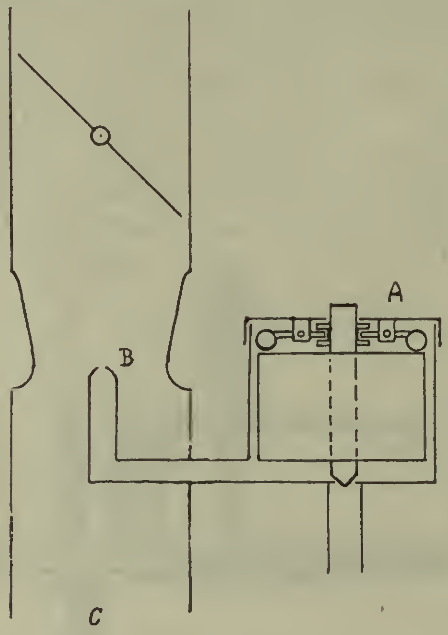


FIG 13

One very successful type of automatic carburetter designed to overcome this difficulty, is shown in Fig. 14 in its vertical and horizontal types.

It will be seen that it is very similar to Fig. 13 with the exception that an extra jet H is provided, termed the auxiliary jet, the other one G being termed the main jet. These jets, the rates of flow of which are shown in Fig. 15 and Fig. 16, supply half the fuel each, the main jet Fig. 15 having a rising curve

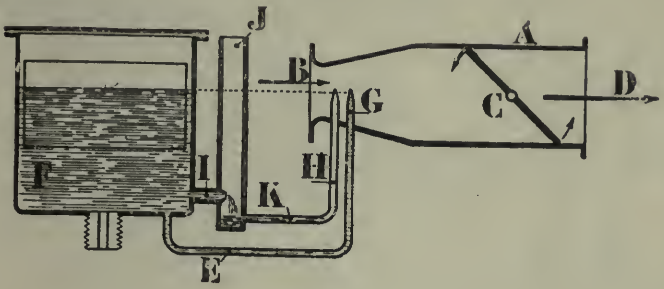


Fig. 14.

which has already been explained, and the auxiliary jet Fig. 16 a falling curve due to it being supplied through the restriction I, the flow of which is not influenced by the depression around the jet H.

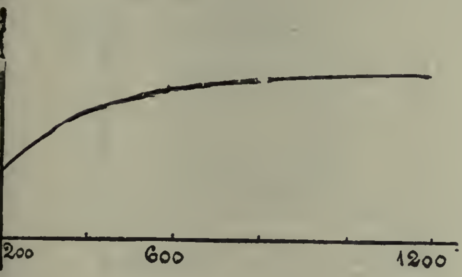


Fig. 15. Main Jet.

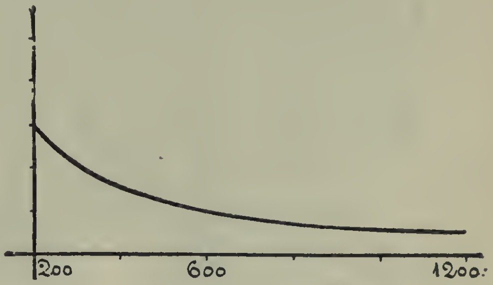


Fig. 16 Auxiliary Jet.

Together these jets correct each other, the mixture remaining practically constant in quality within reasonable limits over a wide range of speed.

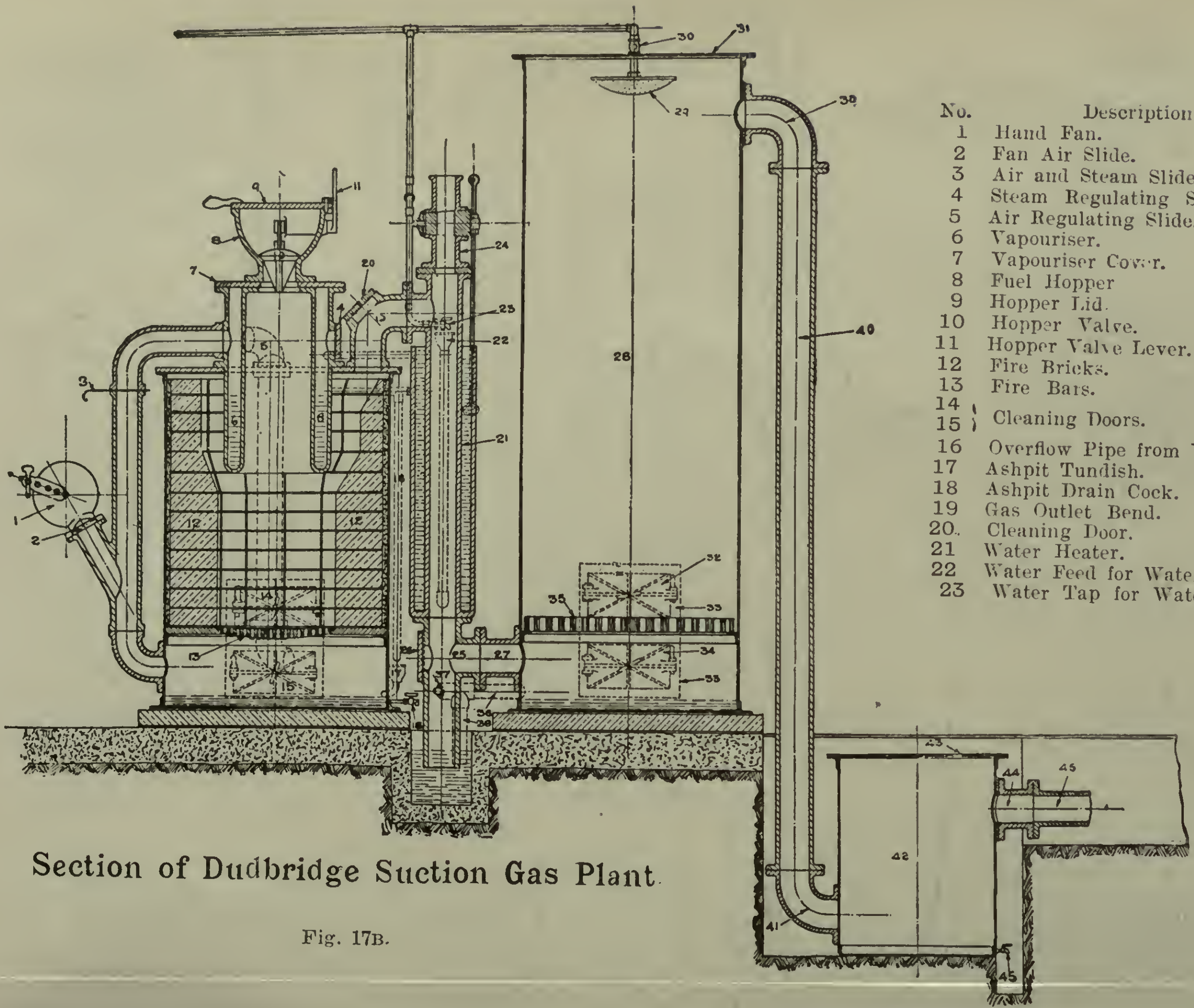
In Figs. 15 and 16 the abscissæ denote engine speeds and the ordinates the mixture strength.

Fig. 17 shows this carburetter in its commercial form with the addition of a slow running jet, a, which is practically a small carburetter in itself.

Gas Producers.—Many gas engines generate their own supply of gas by means of a gas producer Fig. 17B, the engine sucking its supply of air through red hot fuel contained in a generator and then through a scrubber where it is cleaned.

This method of generating gas is very satisfactory, especially from a financial point of view, the cost being very low.

Ignition Jet.—One of the earliest methods of igniting the charge was by means of a gas jet, admitted to the cylinder at the correct moment by a kind of slide valve. The explosion extinguished the jet, which was rekindled by a pilot jet.



Section of Dudbridge Suction Gas Plant.

Fig. 17B.

No.	Description
1	Hand Fan.
2	Fan Air Slide.
3	Air and Steam Slide.
4	Steam Regulating Slide.
5	Air Regulating Slide.
6	Vapouriser.
7	Vapouriser Cover.
8	Fuel Hopper
9	Hopper Lid.
10	Hopper Valve.
11	Hopper Valve Lever.
12	Fire Bricks.
13	Fire Bars.
14 15	Cleaning Doors.
16	Overflow Pipe from Vapouriser.
17	Ashpit Tundish.
18	Ashpit Drain Cock.
19	Gas Outlet Bend.
20.	Cleaning Door.
21	Water Heater.
22	Water Feed for Water Heater.
23	Water Tap for Water Heater.

No.	Description.
24	Gas Escape Cock.
25	Water Seal Pipe.
26	Cleaning Door.
27	Inlet to Scrubber.
28	Scrubber.
29	Water Spray.
30	Water Valve.
31	Scrubber Top.
32	Cleaning Door.
33	Scrubber Door Frames.
34	Scrubber Cleaning Door.
35	Coke Grid.
36 37 38	Water Drain from Scrubber
39	Gas Outlet Bend.
40	Down Pipe to Expansion Box.
41	Expansion Box Inlet Bend.
42	Expansion Box.
43	Expansion Box Cover.
44	Outlet Branch.
45	Drain Top.
46	Gas Pipe to Engine.

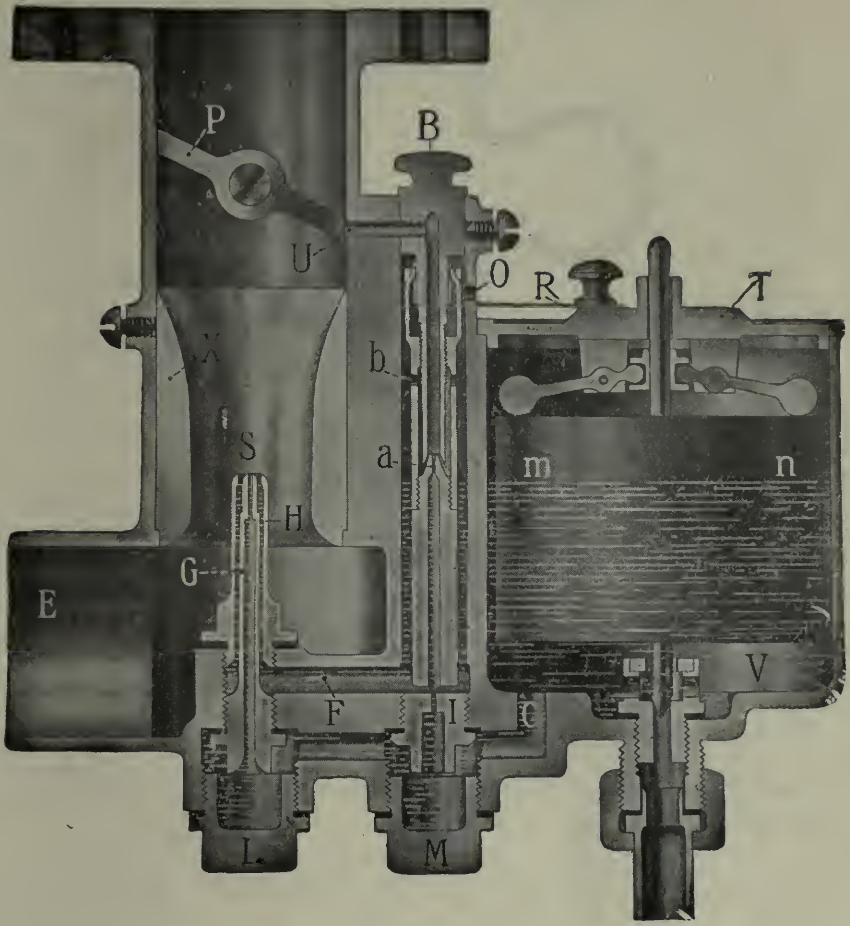


Fig. 17.

Hot Tube.—Jet ignition was soon replaced by the hot tube shown in Fig. 18, which consisted of a tube with a closed end screwed into the combustion chamber and heated by a bunsen, the mixture on compression being forced into the tube and ignited.

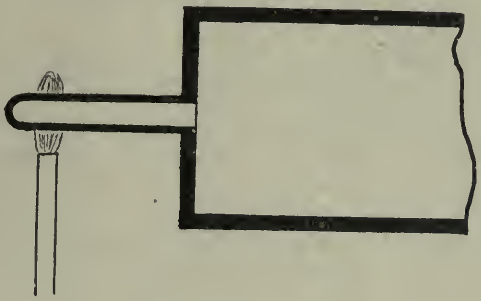


FIG 18

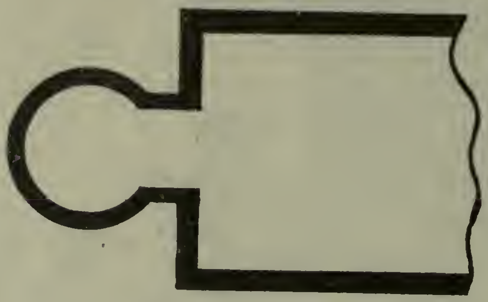


FIG 19

It worked better than at first may be imagined, some slight adjustment of the firing point being obtainable by moving the burner along the tube.

Firing Valve.—Larger engines were fitted with a firing valve which put the tube into communication with the cylinder at the required moment, thus somewhat controlling the firing point.

The early tubes were of platinum and then iron, but later, porcelain proved very satisfactory.

Hot Spot.—The ignition of some types of oil engines is after this style, the bulb acting as a tube as well as a vapouriser, see Fig. 19.

Electric Ignition.—The modern ignition, however, is by means of the electric spark, the various forms being shown by Fig. 20.

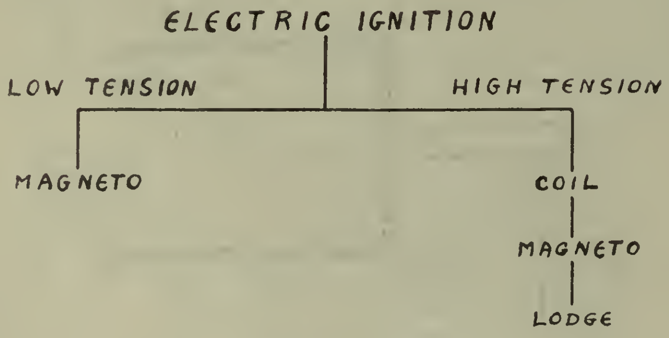


FIG 20

Low Tension.—Ignition by low tension magneto is extensively used for slow speed work, a diagrammatic sketch of the apparatus being shown in Fig. 21.

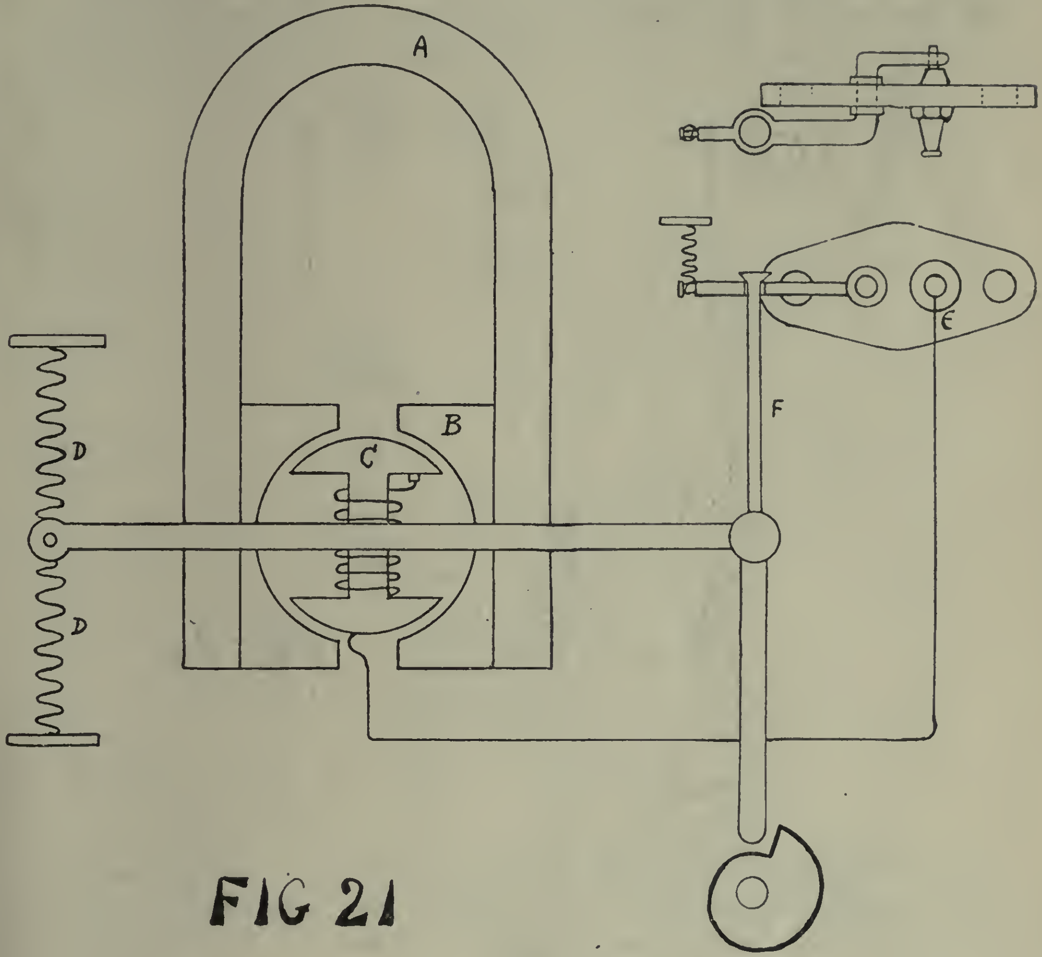


FIG 21

The magneto is of the oscillating type and consists of permanent magnets A, pole pieces B, a shuttle armature wound with fairly thick insulated wire C, and springs D for returning the armature to its position.

At the moment of ignition the cam releases the tappet, giving the armature a strong jerk, this causes a rise of EMF in the winding, one end of which is earthed and the other end led to an insulated button and by a wire to the insulated contact E of the igniter. Immediately after the armature has received its motion the tappet extension F pulls the rocking arm down, thus breaking contact with E and causing the spark.

High Tension Coil.—This type is termed high tension because the spark is not caused by breaking contact, but is given sufficient voltage to enable it to jump a gap at the sparking plug.

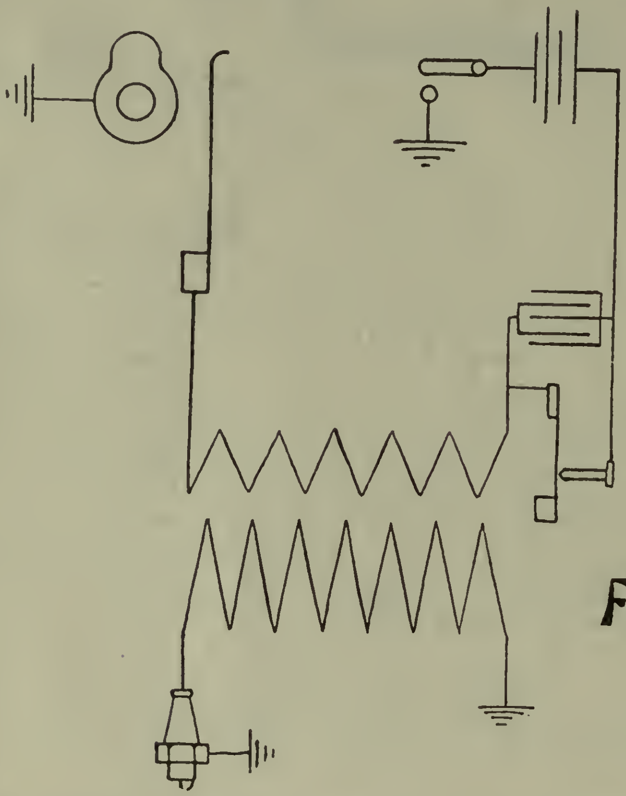


FIG 22

The diagram in Fig. 22 shows one system of connections, the apparatus consisting of a battery, switch, wire contact maker, primary winding a few turns of thick wire, secondary winding many turns of fine wire, condenser, and trembler. The action is as follows:—The switch being closed the contact maker driven from the engine makes contact and completes the primary circuit at the required moment, causing the trembler to vibrate and thus rapidly interrupt the circuit. This generates a high tension current of several thousand volts in the secondary circuit due to the proportion of the windings the current jumping the distance between the sparking plug points.

The practical function of the condenser is to reduce the spark at the trembler contacts which would rapidly destroy them, but incidently it increases the rapidity of demagnetisation of the coil core, thus causing a rapid collapse of the magnetic field which considerably augments the secondary E.M.F.

High Tension Magneto.—This is shown in Fig. 23 and is very similar to the high tension coil with the exception that a battery is not required, the magneto generating its own primary current.

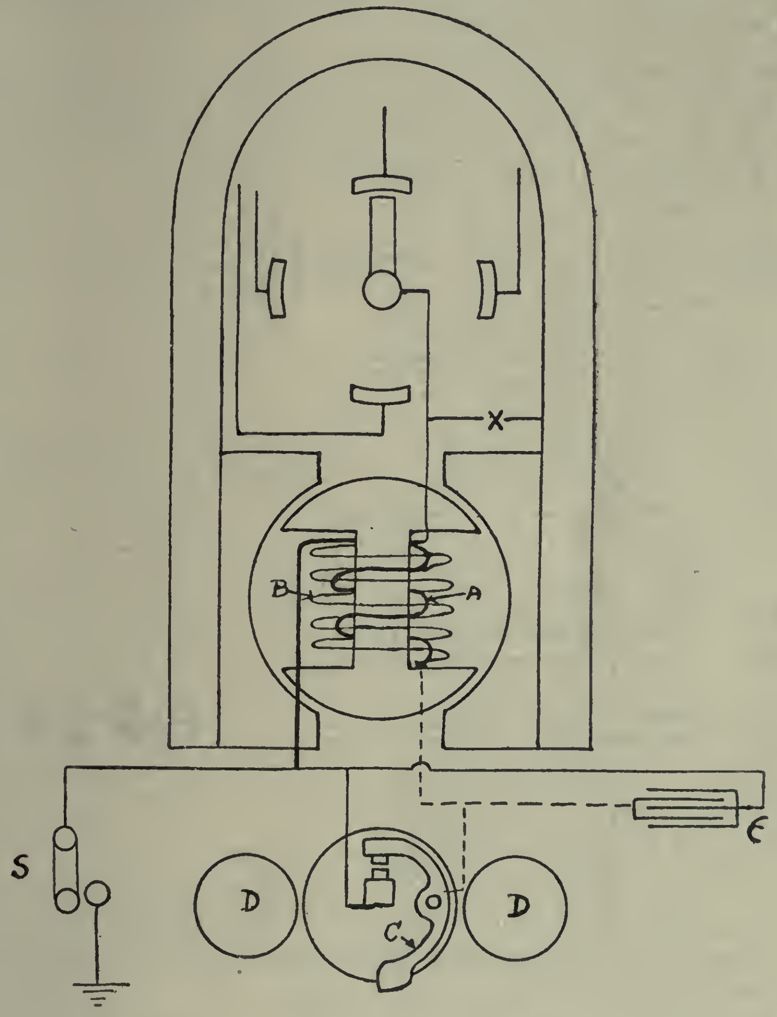


FIG 23

The primary winding A on the armature generates the low tension current, the circuit being broken at the correct moment by the contact breaker C, which is fixed to and revolves with the armature, striking a stationary cam D. This generates the high tension current in the secondary winding B, which is collected by a carbon brush and led to the sparking plugs.

It should be noted that the armature is nearly always arranged to revolve, instead of oscillate, upon this system, and incorporates the condenser E as well as the contact breaker.

A safety gap is provided at X which the spark takes should the plug wires be disconnected.

Lodge.—This ignition is of the coil high tension type and is shown in Fig. 24, the difference being that there are two sparks produced, the spark at A which takes place in the box and is not used and the spark B which is led to the plug.

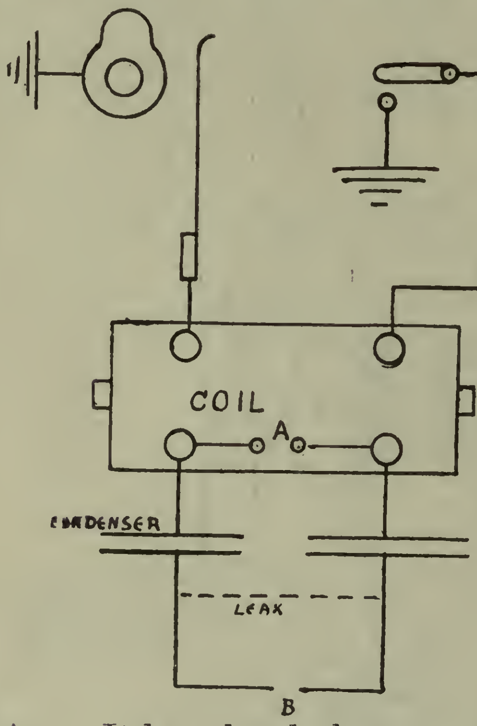


FIG 24

Review.—It has already been stated that the oscillating low tension magneto is used for low speed engines, being very popular with this type.

The high tension coil shown in Fig. 22 has the advantage that it gives a shower of sparks at starting but carries with it the disadvantage of accumulator charging, and the fact that the sparking is not so accurate, due to the uncertain type of contact maker and the lag in the coil.

This system is not much used now, but is sometimes incorporated with the high tension magneto to facilitate starting, this then being termed dual ignition. In America where much attention has been given to this particular type of ignition, it has been redesigned, and now gives the same satisfaction as the high tension magneto, Fig. 25 illustrating the modern version.

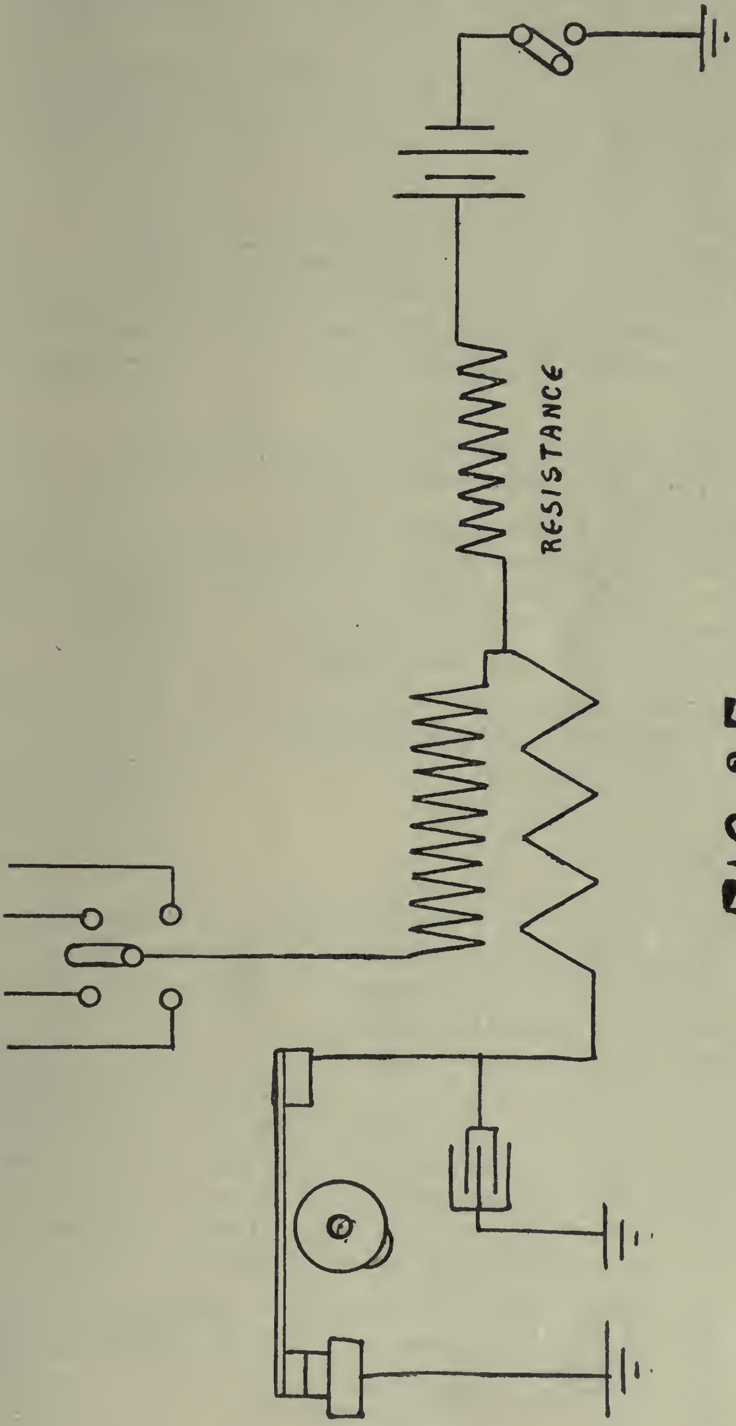


FIG 25

The high tension magneto is used extensively to-day for the ignition of high and moderate speed engines, the firing being accurately timed and it being easily possible to arrange for the ignition of more than one cylinder by the addition of a high tension distributor.

The claims made for the Lodge ignition are that the B spark, as it is called, is of a very detonating character and will prevent carbonisation of the plug, also sparking if the plug should become short circuited.

It has been used upon large stationary engines.

It should be observed that in all engines with any pretence to speed, it is found necessary to make the spark occur earlier than dead centre as the charge takes a perceptible time to ignite, this is done by moving round the contact maker or breaker as the case may be.

In very large cylinders more than one plug is often arranged so as to ignite the charge simultaneously at different points.

The amperage of the current passing at the plug in the high tension type of ignition is very small, due to the considerable step up in voltage taking place in the transforming apparatus, this fact explaining why a considerable shock can be obtained from a plug lead without fatal results.

The sparking voltage between two brass spheres 0·5 ins. in diameter is approximately as follows:—

Distance apart in inches.	Voltage.
0·1	5,000
0·5	15,000
1·0	25,000

The distance apart of the electrodes in the cylinder is about 0·020 in., but it should be remembered that the compression influences the resistance of the gap and also that the facility for the passage of the spark depends upon the area of the electrodes, increasing greatly as one becomes more pointed. In practice they are made of heavy section so as not to cause preignition by becoming incandescent. A difference between coil and magneto ignition with regard to switching off may be mentioned inasmuch that in switching off a coil the primary circuit is broken, whereas with magneto ignition the primary winding is short circuited by earthing through a switch S, Fig. 23.

Fuels.—As previously stated in an earlier part of the paper the internal combustion engine will function satisfactorily upon many different fuels, the study of the characteristics of which becomes quite a complex matter.

Some of the principal points however which interest an engineer are cost, the heating value, and the amount of deposit which they leave in the cylinder.

The cost of course is controlled by many factors and cannot be dealt with here, but an idea of the heating value of different fuels can be obtained from the following tables:—

Gases.	B.T.U.'s per cu. ft. Lower Value.
Natural Gas	900
Coal Gas	691
Water Gas	285
Producer Gas—Coke	135
—Anthracite	145
—Soft Coal	145
Coke Oven Gas	545
Blast Furnace Gas	100
*Acetylene Gas	1,480

Oils.	S.G. at 32° C.	B.T.U.'s per lb. Lower Value.
Crude average	0·870	19,100
Solar	0·870	
Petrol	0·680	19,300
,,	0·720	18,500
,,	0·760	18,300
Kerosene, American	0·800	18,500
†Alcohol	0·810	11,600
Benzol	0·880	18,200

Mixtures.	Air required for Combustion cu. f. per cu. ft. of Gas.
Natural Gas	9·0
Coal Gas	5·25
Producer Gas—Coke	1·00
,, ,, Anthracite	1·15
,, ,, Soft Coal	1·25
Coke Oven Gas	5·00
Blast Furnace Gas	0·70
Petrol, maximum power	14·00
,, ,, efficiency	16·50
Alcohol	11·00
Benzol, maximum power	16·20
,, ,, efficiency	20·10
Acetylene	12·00

* 1 lb. Carbide = 5 cu. ft. of Gas.

† Can be made from Potatoes, Rye, Starch, etc., Denaturised by the addition of Wood Alcohol.

With regard to the deposit question it may be said that all fuels are offenders in this respect, the combustion chamber having to be cleaned out periodically, engines running upon poor fuels collecting a tar-like substance upon the valves, which, unless cleaned off, prevents their proper functioning. With the highest grade fuels such as petrol, the intervals between cleaning are of course much longer, the deposit in this case being harder, the limit of time being reached when knocking takes place, due to the coating of carbon, reducing the size of the combustion chamber, or a piece of carbon upon one part assuming such proportions that it causes preignition, due to remaining incandescent.

An accurate idea of the conditions prevailing in the cylinder, besides those observed by the indicator, can be obtained by an exhaust gas analysis. This is easily done, the apparatus consisting of a frame holding pipettes containing solutions of—

Caustic Potash to absorb CO_2.

Alkaline Pyrogallol to absorb O.

Acid Cuprous Chloride to absorb CO.

The gas is brought into contact with the solutions, the amount of these constituents indicating the efficiency of the explosion, an excess of O for instance indicating weak, and an excess of CO rich, mixtures.

Engine Speeds.—Engines can of course be constructed to run at speeds suiting different purposes, but on an average, speeds are higher than those used in steam practice, varying from 90 r.p.m. for marine engines to 3,500 r.p.m. for automobile racing engines, it being not by any means an easy proposition to control the position of the knee on the speed power curve.

Weights.—Under this heading information is rather difficult to collect, but it may be stated that the engine shown in Figs. 26 and 27, built by the firm which constructed the engines for the airship R34, weighs less than 3lbs. per horse power, and was used during the war in 55ft. boats for chasing submarines.

Aero engines of a similar type often fall below 2lb. per horse power, whilst at the other end of the scale we find two cycle Diesel engines at 160lbs. per horse power and four cycle Diesel engines at 260lbs. per horse power. It should be noted that due to the heavy strains set up by the high temperatures and method of working, Diesel engines are of very robust construction.

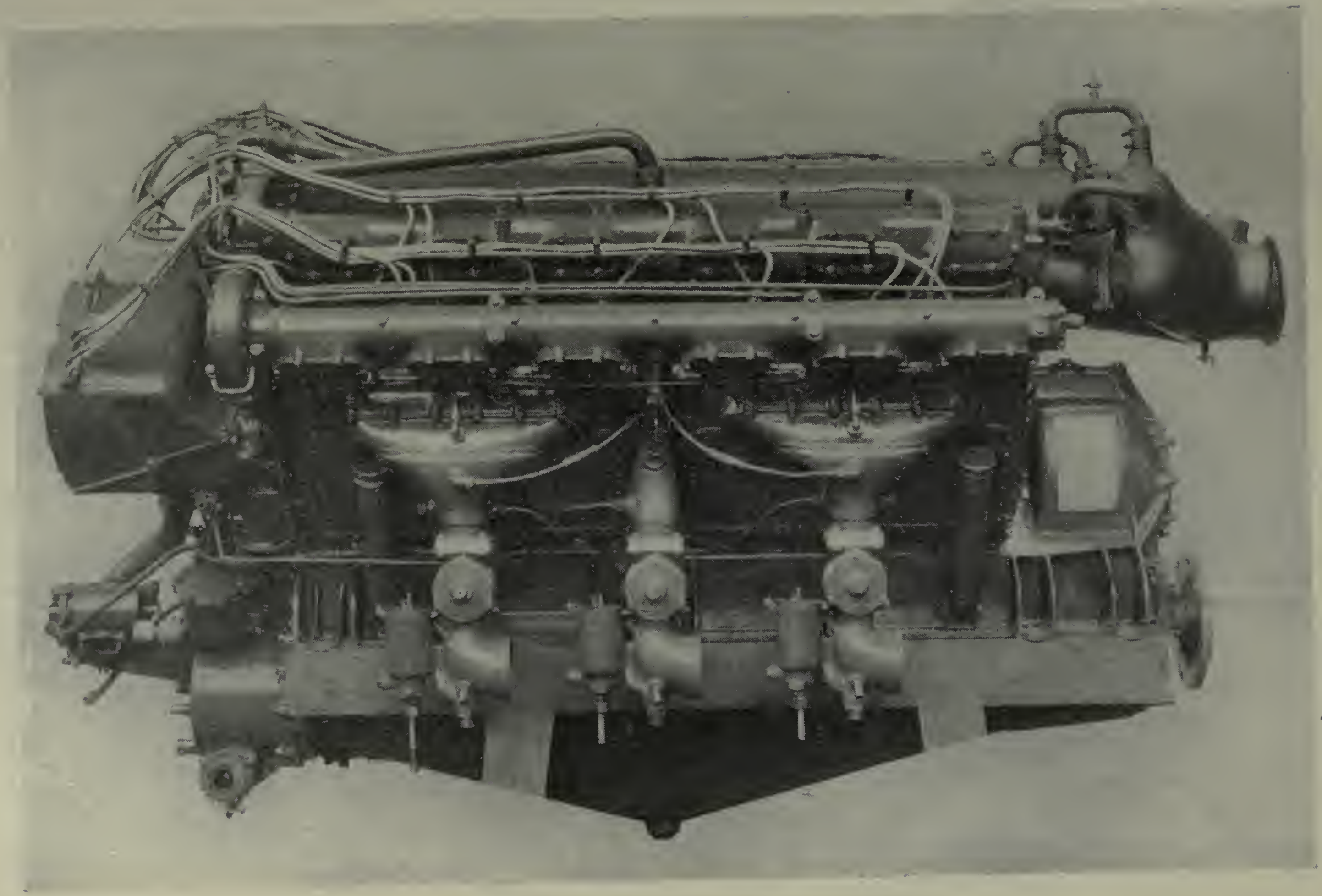


Fig. 26.

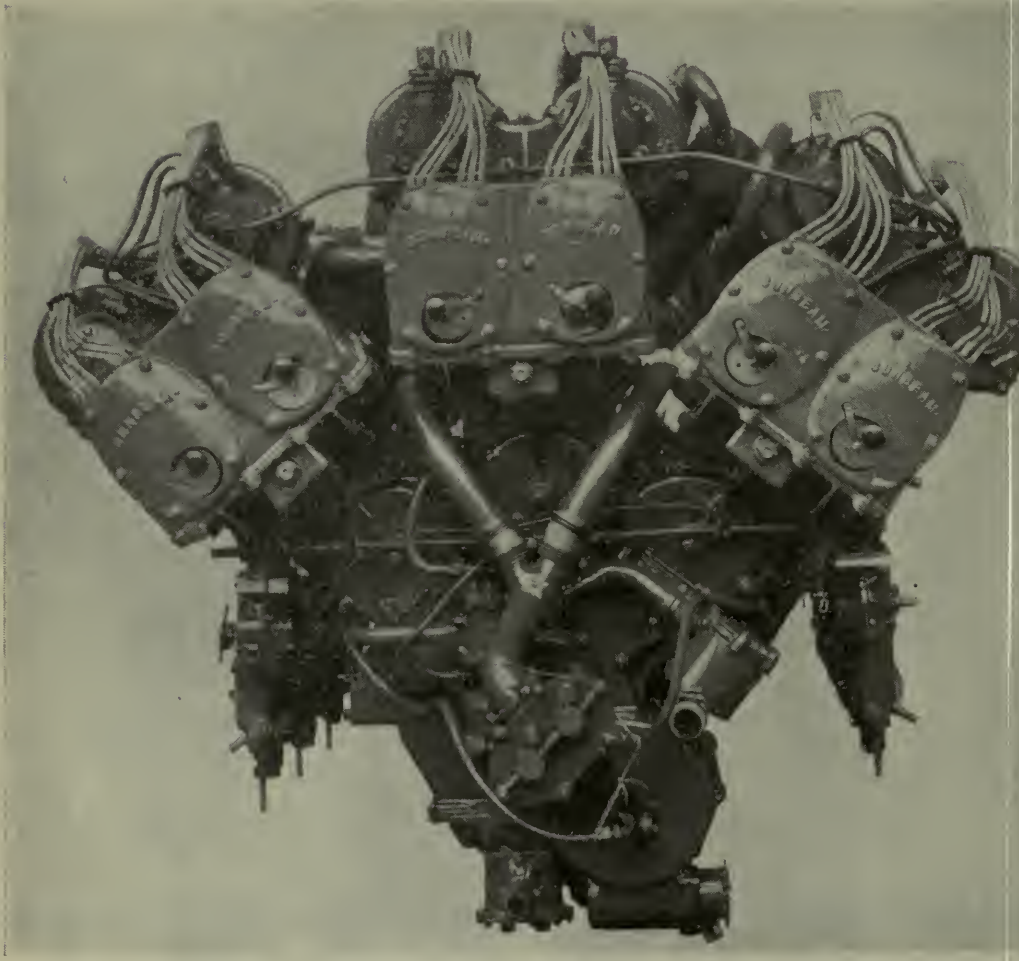


Fig. 27.

Valves.—The slide type of valve previously mentioned, was found to be unsatisfactory and the poppet valve soon came into universal use, being lifted by a cam operated by gearing and returned to its seating by means of a coiled spring. As cast iron was found to withstand the heat well they are often made by screwing a steel stem into a cast iron head, but with high speeds and compression pressures where valves often operate for long periods at a cherry red heat, it has been found necessary to construct them of special steels which resists the warping and burning action, a tungsten type of steel being often used.

For high speed work particular attention has to be paid to the shape of the gas passages in order to give the engines a good volumetric efficiency or free breathing capacity, the valve in head system Fig. 28 having important advantages over the T headed system shown in Fig. 29.

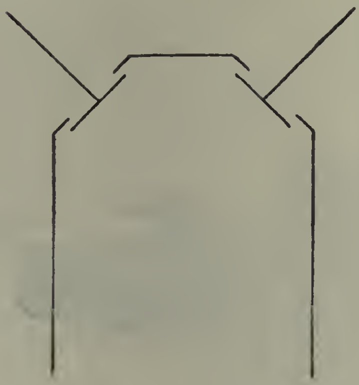


FIG 28

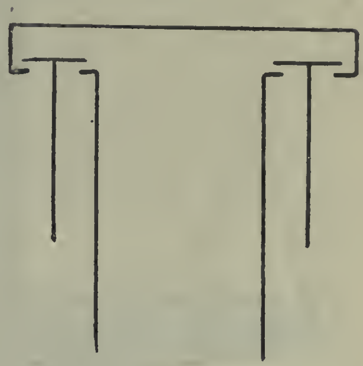


FIG 29

It must not be forgotten, however, that in a mixture, in which no eddying or whirling is taking place, a far longer period must elapse after ignition before the maximum explosion pressure is reached, than could be possibly allowed in any practicable engine. There is therefore a limit in every engine past which it is unwise to carry the free breathing question, if satisfactory running is to be obtained.

The arrangement and design of the valves and passages influences the valve setting, it being found necessary to take advantage of the gas inertia and thus not open and close the valves upon the dead centres.

Valve settings differ considerably, some makers attempting to get rid of the dead gas in the unswept portion of the cylinder by lapping the valves, *i.e.*, allowing the inlet valve to open a few degrees before the exhaust valves closes, so taking advantage of the inertia of the exhaust gas to draw the new charge across the piston top.

Other types of valves have been introduced from time to time such as the disc, plug and piston types, but the only one attaining any degree of popularity, and this only in one field, is the sleeve valve, Fig. 30.

The sleeves are reciprocated by small eccentrics, the ports registering at the desired points in the stroke.

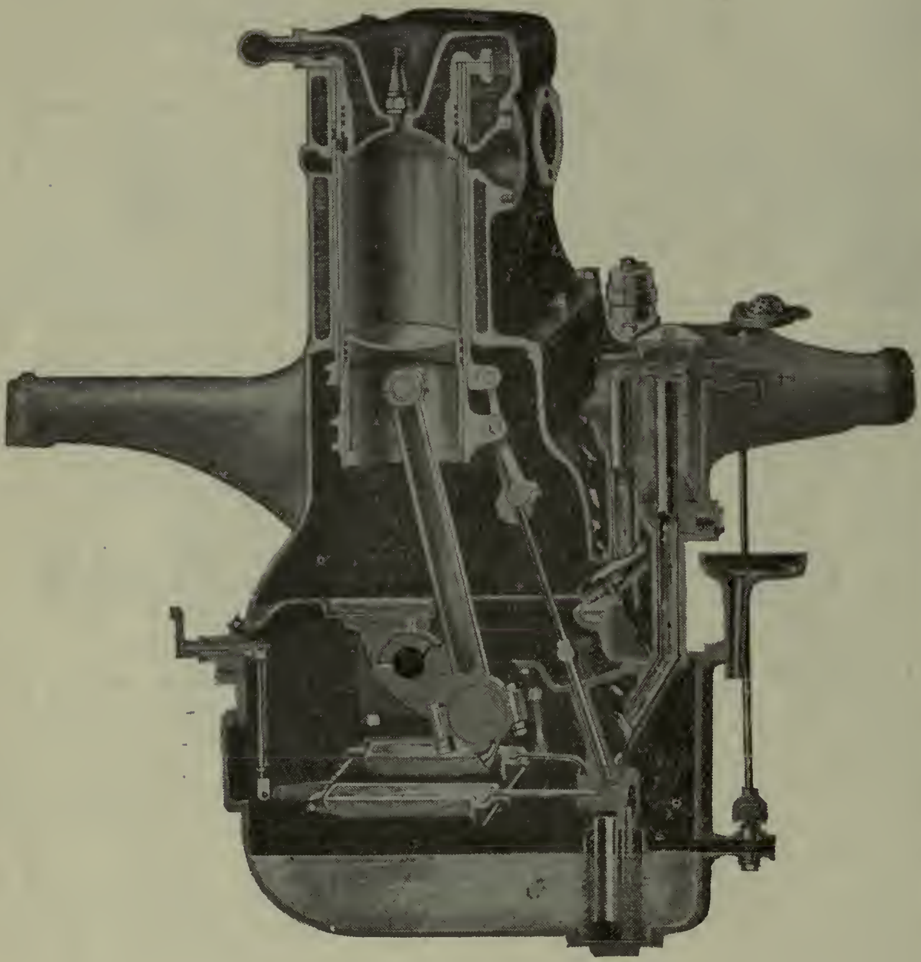


Fig. 30.

Compression Pressures.—It is found an advantage upon theoretical grounds to use as high a compression pressure as possible, the limit being reached when spontaneous combustion takes place, upon the compression stroke, due to the heating up of the charge.

This critical pressure varies with different fuels, being low for fuels containing a large hydrogen content. The general design of an engine also influences the pressure permissible, as will be easily seen, engines with ample water jackets and no overheated local spots in the cylinder permitting the use of a higher compression ratio.

THE DEVELOPMENT OF THE INTERNAL COMBUSTION ENGINE.

The volumetric efficiency of an engine, spoken of under the heading of Valves, is represented by the equation,—

$$\frac{X}{Vp}$$

Where—

X = Charge actually inducted.

Vp – Volume swept out by the piston.

The compression ratio which plays such a vital point in engine design is given by—

$$\frac{V+Vp}{V}$$

Where—

V = Volume of compression space.

In settling a suitable compression ratio there is not only the critical spontaneous compression pressure to contend with, but also the rate of flame propagation throughout the mixture, which varies with different fuels. For instance, acetylene gas and also paraffin mixtures explode with much violence and often cause severe knocking, which has led some makers to introduce water with the mixture, by means of a drip, in order to tone down the explosion, when the engine is under full load.

The compression of the charge is approximately adiabatic, the expression for which is

$$PV^n = \text{Constant.}$$

The compression pressure is given by the formula

$$P_c = P\left(\frac{V + V_p}{V}\right)^n$$

Where,—

Pc = Compression pressure absolute.

P = Pressure before compression.

As the expression $\frac{V + V_p}{V}$ is the compression ratio which can be represented by R, the compression in terms of it is given by,—

$$P_c = PR^n$$

The pressure P of the charge before compression is governed by the volumetric efficiency, it being in well designed engines about 14·2lbs. per square inch at low speeds and 11 to 13·75lbs. per square inch at normal speeds, the former figure being for a moderately well designed high speed automobile type engine, and the latter for a well designed slow speed gas engine.

Other everyday examples fall somewhere between 12·7 and 13·5, 13·1 being a fair average.

The value of the exponent N is in the region of 1·3; a fairly accurate representation of its value being 1·27 for the compression and 1·32 for the expansion line.

The following table gives some of the successful pressures used:—

Fuel.	Engine.	Pressure lbs. per sq. inch gauge.
Paraffin... ...	Four cycle boat	65
Petrol	Two cycle automobile ...	70
Petrol	Four cycle automobile ...	85
Illuminating gas	Four cycle gas	80
Natural gas ...	Four cycle gas	100
Producer gas ...	Four cycle gas	130
Blast furnace gas	Four and two cycle	160
Oil	Semi-Diesel boat	150-210
Alcohol... ...	Automobile type	200
Diesel	Two Cycle Marine	450 and upwards
Petrol	Aero engine and racing automobile	130

The following table is also interesting as it gives an idea of the maximum explosion and mean effective pressures prevailing:—

Engine.	Explosion pressure. gauge.	M.E.P.
Gas Engine, four cycle ...	450	—
Diesel Engine, four cycle ...	685	151
Diesel Marine, two cycle ...	650	110
Automobile	300	84
Automobile	295	81
Aero Engine, air cooled ...	432	90
Aero Engine, water cooled ...	454	93

Efficiency.—A great point in favour of the adoption of internal combustion engines is the high thermal efficiency obtainable, the following table giving the approximate growth in efficiency of the various prime movers from early times to the present date.

Prime Mover.	Indicated Thermal Efficiency %.
Boulton and Watt Condensing	3·8
Lenoir Gas Engine	4·0
Cornish Engine	9·0
Four Stroke Cycle, on inception, Gas ...	16·0
Triple Expansion Condensing	18·0
Oil Engine	22·0
Parson's Turbine	23·0
Petrol Engine, Automobile	25·0
Gas Engine	28·0
Diesel Two Cycle	33·0
Diesel Four Cycle	38·0
Still Engine (said to be)	44·0

The losses in the internal combustion engine are variously estimated by the different authorities, but the following is a fair average :—

Loss.	%
In Exhaust	52·5
In Water Jacket	15·3
Mechanical Loss	4·7
Power at Crankshaft	27·5
	100·0

Design.—With regard to the design of this type of engine it may be said that the subject is now of a more settled nature compared with a few years ago when engines were built by rule of thumb methods.

Results are now foretold with considerable exactitude, all stresses being fairly accurately computable, a great amount of research work having overcome numerous difficulties.

A small amount of trouble is still met with in connection with the heat gradient in cylinder walls and pistons, resulting in the occasional cracking of these parts from unequal expansion, this being confined principally to the larger sizes.

Many makers are now constructing engines with cylinders set desaxialy, *i.e.*, setting the cylinder not directly over the crankshaft, thus equalising the pressure upon the cylinder walls, the amount of setting being reasonable, to avoid complications in other directions.

Balancing has also received considerable attention since it became such an important factor in high-speed engine design, many makers carrying this out with great care, installing apparatus for accurate static and dynamic balancing.

This has assumed such importance with certain makers of automobile engines that some of them fit a device for damping the oscillatory whip of crankshafts which is very liable to occur with six cylinder engines.

More attention is also being paid to the question of metallurgy, the heat treatment of steels being very much to the fore, all up-to-date firms considering this a very important subject, the department being under the direct control of the works laboratory.

Some firms, however, as in all other branches of engineering, are still without even a microscope or testing machine, purchasing their materials in the ordinary way and trusting the vendor implicitly, the whole sum total of the heat treatment given being the case-hardening of certain parts; this is not confined to the makers of one type of engine but applies to makers in practically every field.

It may be argued that failures due to bad material are few and far between, but this only indicates that the design is bad, the factor of safety being obviously too high. Makers who possess a laboratory, in nearly every case illustrate it in their catalogue, showing that the importance of the fact is realised.

Interchangeability is also vastly better than it was, this, of course, being brought about by a free use of the limit system, but, nevertheless, some firms seem to be only just wakening up at the present time, and studying the advantages derivable from its use. The drawing offices of many otherwise up-to-date firms do not seem to have moved with the times, the obviously correct method of designing upon an economical production basis being entirely unknown. This is brought about by the fact that the designers have had very little shop experience in the true sense of the word; so long as an engine can be built to run satisfactorily, they seem to remain entirely oblivious of the fact that it is possible to cut off an immense slice of the manufacturing costs by suitably arranging the design with a view to ease of machining and fitting. This art can only be acquired by spend-ing a number of years in the shops, one, two or three or even five years' experience is not enough, it must be considerably more to be of any consequence, as the members of this Institute will agree.

Engines and machines of all descriptions are being built in this country to-day by different firms, without apparently the slightest consideration being given to this question. This lack of workshop experience also reflects itself in such points as accessibility, and an inspection of the blueprints issued by some firms will indicate that even in the method of numbering drawings unnecessarily, complicated systems are in vogue which waste much valuable time even when a draughtsman gets to thoroughly understand them.

The previously mentioned limit gauge system also needs to be applied with considerable intelligence or it will be found to be expensive; some draughtsmen having no idea what effect a few thousandths of an inch play will have upon the correct functioning of certain parts, and therefore do not know whether to apply a wide or narrow limit, thus hindering speed of production.

There is a very valuable table of limits broadly published, but this as its originators inform people, is not intended to be applied to all branches of engineering, and in spite of this, it is applied almost universally without any attention to this point. The remedy is for each drawing office to modify the table to suit its own particular work, which is not a difficult matter. Volumes could be written upon the subject of design for economical production, but sufficient has been said to indicate that the present method of installing tool specialists in the works to cut down manufacturing costs, after the design is completed, is not enough, as little can then be done compared with what can be accomplished if the subject be tackled in its proper place, *i.e.*, in the drawing office.

Present and Future Developments.—The prime mover with which the present paper has attempted to deal in a general way has now been extant for over a hundred years, during which time, great strides have been made in its design.

The whole subject progressed very slowly at first, evidently due to the fact that few engineers were really interested in its welfare, but when the right line of construction was reached, great steps were at once made in the development.

This rapid progress continued for some years, each new discovery opening up new fields, until, as in the case of all other prime movers the design as we know it to-day, seems very unlikely to be radically changed, the principles now being well understood.

Great speculation naturally arises in the minds of scientifically inclined people as to what form the next great advance will take, a lull apparently having fallen over the evolution of new prime movers. Supercharging, *i.e.*, forcing in an extra quantity of mixture or air at the end of the induction stroke has been accomplished with a little success, but nothing very revolutionary has transpired, the engine becoming complicated by the addition of such items as pump cylinders.

Suggestions have been made from time to time that a great advance in thermal efficiency might be obtained by combining the steam and the internal combustion engine. This idea occurred to the writer several years ago when a student, and a simple design was got out, but the simple design eventually assumed such a complicated form that the idea was given up for the time being.

The scheme was mentioned to Professor Spooner, which brought to light the information that one of Professor Perry's students had attempted the same thing some years previously. A successful engine of this form has now been evolved by Mr. Still, and judging by the result of tests upon the specimen constructed, is likely to justify much of the confidence reposed in it.

The chief problem, however, which has exercised the minds of designing engineers for a long time, the writer not being excepted, is the construction of a successful gas turbine, it being considered that this is the next and final step in the development of devices for internal combustion. Considering that this was the first engine known, being described in its steam form by Hero in the year 120 B.C., it is very strange that development has not proceeded more upon these lines.

The problems which have had to be surmounted in its steam prototype have shown that the question is by no means as simple as one would imagine considering its elementary form; and when it is considered from an internal combustion standpoint, the difficulties are seen to be enormously magnified. The great heat generated is of course the principal obstacle, it being very difficult to construct a machine which will even run for any length of time, but now that refractories are being studied with greater vigour, there should be more hope upon this point. Two suggested arrangements are shown in Figs. 31 and 32.

THE DEVELOPMENT OF THE INTERNAL COMBUSTION ENGINE.

Fig. 31 consists of a chamber of refractory material in which the combustion of the fuel takes place, the gases impinging upon buckets on a wheel, Pelton or De Laval style.

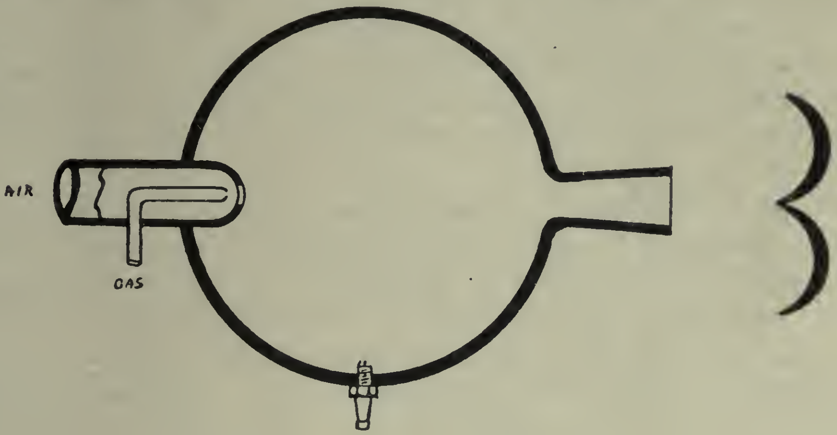


FIG 31

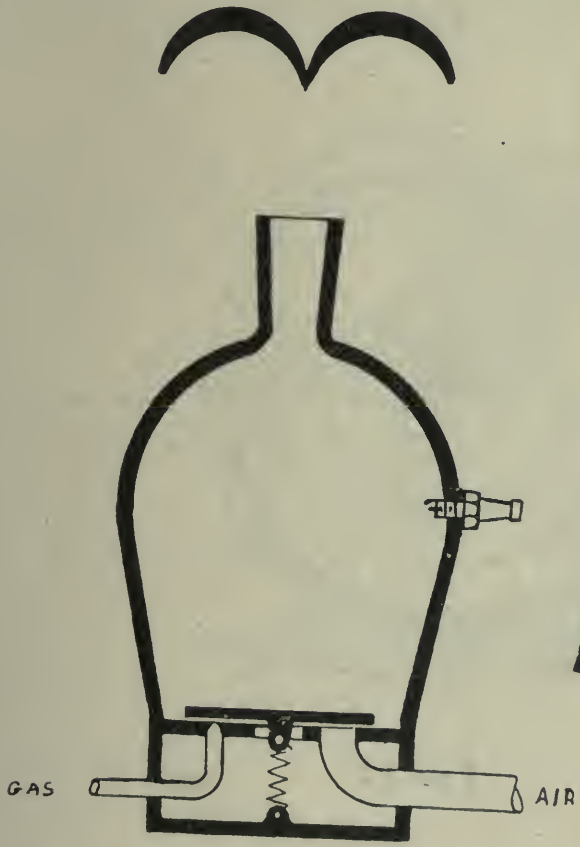


FIG 32

Fig. 32 is of a pulsating type, the action being obtained as follows:—

The chamber being charged and fired, the gas rushes out of the jet, overrunning itself by inertia and drawing another charge of mixture after it through the flap valve, the action being rapidly repeated. It will be seen that this is very analagous to the action which takes place in a gas burner when turned very low.

The heat generated in these devices is of course very great, and the materials at present at our disposal do not inspire much confidence in development in this direction, the combustion being so continuous that no chance is given for cooling.

In the ordinary type of high-speed engine the explosions, although very rapid, are intermittent, which allows of the combustion chamber walls being kept at a reasonable temperature, the incoming cool gas being of great use in this direction. Working upon these lines the writer attempted some years ago to evolve a design, one of the results being shown in sectional elevation in Fig. 33, Fig. 34 being the inlet side, Fig. 35 a view on the exhaust side, and Fig. 36 a plan in part section.

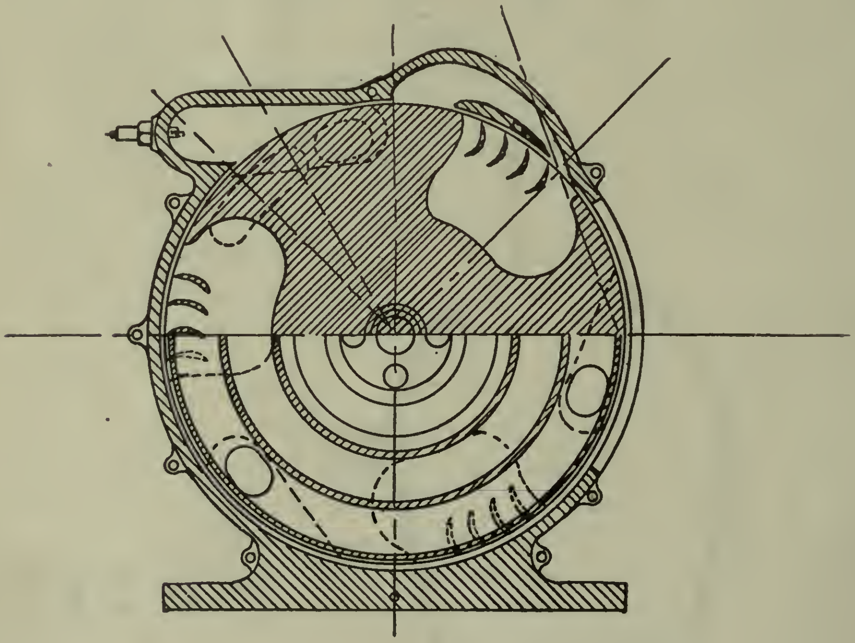
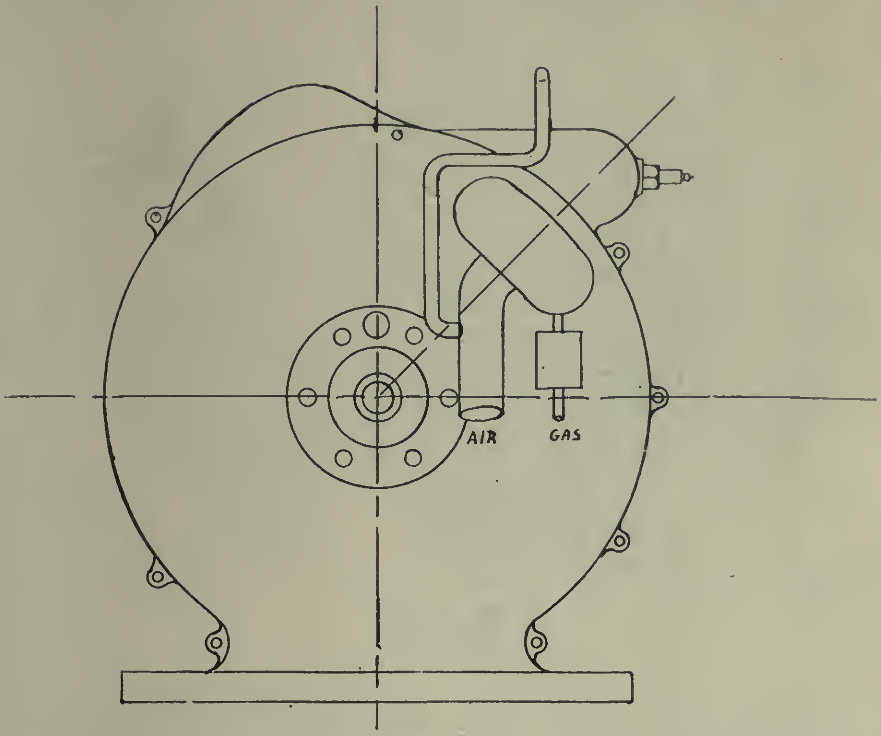


FIG 33



.FIG 34

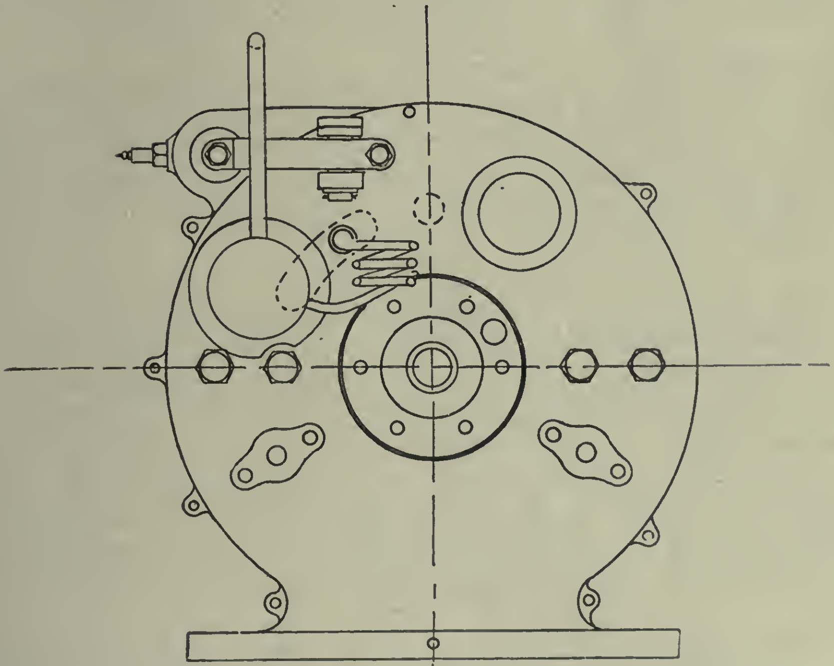


FIG 35

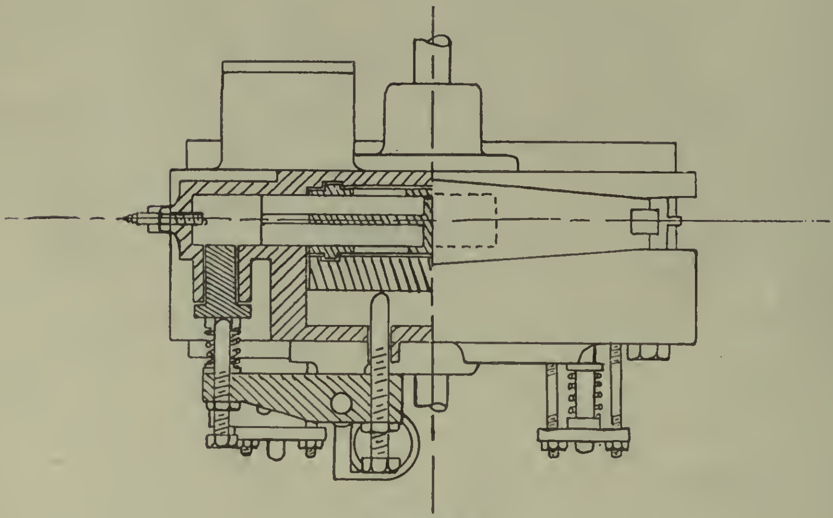


FIG 36

The gas was exploded in a chamber forming part of the casing, the resulting explosion doing work by impinging upon the chamber in the rotor from which it escaped being returned again to the rotor vanes by means of a jet, the following charge under pressure scavenging the chamber and refilling it.

Such difficulties as holding the ported rings up to the rotor with just the correct pressure, were surmounted by balancing the explosion pressure by means of a piston and fulcrum, Fig. 36, the proportions being such that the pressure tending to force the ring from the rotor face was exactly balanced by the pressure upon the piston.

It will be seen that the foregoing arrangements involve the use of turbo compressors, the efficiencies of which are at present low, and even if a compressor of reasonable efficiency was forthcoming there is no direct prospect of high combined efficiencies being obtainable. A combined steam and gas turbine, however, may do much towards solving the problem, and the writer already sees great prospects in this direction, the resulting machine even if its efficiency was not such as to render existing prime movers obsolete, being undoubtedly of some utility.

In conclusion, the writer begs to tender his thanks to makers who have kindly loaned blocks of engines for the purpose of illustration.

TOLERANCES IN STANDARD HOLES.

(2 Grades.)

	NOMINAL DIAMETERS.	Up to $\frac{1}{2}$″	$\frac{9}{16}$″-1″	$1\frac{1}{16}$″-2″	$2\frac{1}{16}$″-3″	$3\frac{1}{16}$″-4″	$4\frac{1}{16}$″-5″	$5\frac{1}{16}$″-6″
Class **A**	High Limit	+·00025	+·00050	+·00075	+·00100	+·00100	+·00100	+·00150
	Low ,,	—·00025	—·00025	—·00025	—·00050	—·00050	—·00050	— 00050
	Tolerance	·00050	·00075	·00100	·00150	·00150	·00150	·00200
Class **B**	High Limit	+·00050	+·00075	+·00100	+·00125	+·00150	+·00175	+·00200
	Low ,,	—·00050	—·00050	—·00050	—·00075	—·00075	—·00075	—·00100
	Tolerance	·00100	·00125	·00150	·00200	·00225	·00250	·00300

ALLOWANCES FOR VARIOUS FITS.

FORCE FITS.

	NOMINAL DIAMETERS.	Up to $\frac{1}{2}$″	$\frac{9}{16}$″-1″	$1\frac{1}{16}$″-2″	$2\frac{1}{16}$″-3″	$3\frac{1}{16}$″-4″	$4\frac{1}{16}$″-5″	$5\frac{1}{16}$″ 6″
Class **F**	High Limit	+·00100	+·00200	+·00400	+·00600	+·00800	+·01000	+·01200
	Low ,,	+·00050	+·00150	+·00300	+·00450	+·00600	+·00800	+·01000
	Tolerance	·00050	·00050	·00100	·00150	·00200	·00200	·00200

DRIVING FITS.

	NOMINAL DIAMETERS.	Up to $\frac{1}{2}$″	$\frac{9}{16}$″-1″	$1\frac{1}{16}$″-2″	$2\frac{1}{16}$″-3″	$3\frac{1}{16}$″-4″	$4\frac{1}{16}$″-5″	$5\frac{1}{16}$″-6″
Class **D**	High Limit	+·00050	+·00100	+·00150	+·00250	+·00300	+·00350	+·00400
	Low ,,	+·00025	+·00075	+·00100	+·00150	+·00200	+·00250	+·00300
	Tolerance	·00025	·00025	·00050	·00100	·00100	·00100	·00100

PUSH FITS.

	NOMINAL DIAMETERS.	Up to $\frac{1}{2}$″	$\frac{9}{16}$″-1″	$1\frac{1}{16}$″-2″	$2\frac{1}{16}$″-3″	$3\frac{1}{16}$″-4″	$4\frac{1}{16}$″-5″	$5\frac{1}{16}$″-6″
Class **P**	High Limit	—·00025	—·00025	—·00025	—·0005	—·0005	—·0005	—·0005
	Low ,,	—·00075	—·00075	—·00075	—·0010	—·0010	—·0010	—·0010
	Tolerance	·0005	·0005	·0005	·0005	·0005	·0005	·0005

RUNNING FITS (3 Grades).

	NOMINAL DIAMETERS.	Up to $\frac{1}{2}$″	$\frac{9}{16}$″-1″	$1\frac{1}{16}$″-2″	$2\frac{1}{16}$″-3″	$3\frac{1}{16}$″-4″	$4\frac{1}{16}$″-5″	$5\frac{1}{16}$″-6″
Class **X**	High Limit	—·00100	—·00125	—·00175	—·00200	—·00250	..·00300	—·00350
	Low ,,	—·000 0	—·00275	—·00350	—·00425	—·00500	—·00575	—·00650
	Tolerance	·00100	·00150	·00175	·00225	·00250	·00275	·00300
Class **Y**	High Limit	—·00075	— 00100	—·00125	—·00150	—·00200	—·00225	—·00250
	Low ,,	—·00125	—·00200	—·00250	—·00300	—·00350	—·00400	—·00450
	Tolerance	·00050	·00100	·00125	·00150	·00150	·00175	·00200
Class **Z**	High Limit	—·00050	— 00075	—·00075	—·00100	—·00100	—·00125	—·00125
	Low ,,	—·00075	—·00125	—·00150	—·00200	—·00225	—·00250	—·00275
	Tolerance	·00025	·00050	·00075	·00100	·00125	·00125	·00150

THE DEVELOPMENT OF THE INTERNAL COMBUSTION ENGINE.

TOLERANCES IN STANDARD HOLES.

(2 Grades.)

	Nominal Diameters.	6$\frac{1}{16}$"-7"	7$\frac{1}{16}$"-8"	8$\frac{1}{16}$"-9"	9$\frac{1}{16}$"-10"	10$\frac{1}{16}$"-11"	11$\frac{1}{16}$"-12"
Class **A**	High Limit	+·00150	+·00175	+·00175	+·00175	—·00200	+·00200
	Low ,,	—·00075	—·00075	—·00100	—·00100	—·00100	—·00100
	Tolerance	·00225	·00250	·00275	·00275	·00300	·00300
Class **B**	High Limit	+·00225	+·00225	+·00250	+·00250	+·00275	+·00275
	Low ,,	—·00100	—·00125	—·00125	—·00125	—·00125	—·00150
	Tolerance	·00325	·00350	·00375	·00375	·00400	·00425

ALLOWANCES FOR VARIOUS FITS.

FORCE FITS

	Nominal Diameters.	6$\frac{1}{16}$"-7"	7$\frac{1}{16}$"-8"	8$\frac{1}{16}$"-9"	9$\frac{1}{16}$"-10"	10$\frac{1}{16}$"-11"	11$\frac{1}{16}$"-12"
Class **F**	High Limit	+·014	+·016	+·018	+·020	+·022	+·024
	Low ,,	+·012	+·014	+·016	+·018	+.020	+·022
	Tolerance	·002	·002	·002	·002	·o 2	·002

DRIVING FITS.

	Nominal Diameters.	6$\frac{1}{16}$"-7"	7$\frac{1}{16}$"-8"	8$\frac{1}{16}$"-9"	9$\frac{1}{16}$"-10"	10$\frac{1}{16}$"-11"	11$\frac{1}{16}$"-12"
Class **D**	High Limit	+·00450	+·00500	+·00550	+·00600	+·00650	+·00700
	Low ,,	+·00300	+·00350	+·00400	+·00450	+·00450	+·00500
	Tolerance	·00150	·00150	·00150	·00150	·00200	·00200

PUSH FITS.

	Nominal Diameters.	6$\frac{1}{16}$"-7"	7$\frac{1}{16}$"-8"	8$\frac{7}{16}$"-9"	9$\frac{1}{16}$"-10"	10$\frac{1}{16}$"-11"	11$\frac{1}{16}$"-12"
Class **P**	High Limit	—·00050	—·00050	—·00050	—·00075	—·00075	—·00075
	Low ,,	—·00125	—·00150	—·00150	—·00200	—·00200	—·00200
	Tolerance	·00075	·00100	·00100	·00125	·00125	·00125

RUNNING FITS.

	Nominal Diameters.	6$\frac{1}{16}$"-7"	7$\frac{1}{16}$"-8"	8$\frac{1}{16}$"-9"	9$\frac{1}{16}$"-10"	10$\frac{1}{16}$"-11"	11$\frac{1}{16}$"-12"
Class **X**	High Limit	—·00350	—·00350	—·00375	—·00400	—·00400	—·00425
	Low ,,	—·00675	—·00700	—·00750	—·00800	—·00825	—·00850
	Tolerance	·00325	·00350	·00375	·00400	·00425	·00425
Class **Y**	High Limit	—·00275	—·00275	—·00300	—·00325	—·00325	—·00350
	Low ,,	—·00475	—·00500	—·00550	—·00575	—·00600	—·00625
	Tolerance	·00200	·00225	·00250	·00250	·00275	·00275
Class **Z**	High Limit	—·00125	—·00150	—·00150	—·00150	—·00175	—·00175
	Low ,,	—·00275	—·00300	—·00300	—·00325	—·00350	—·00350
	Tolerance	·00150	·00150	·00150	·00175	·00175	·00175

Fig. 37 Newall Limit Table.

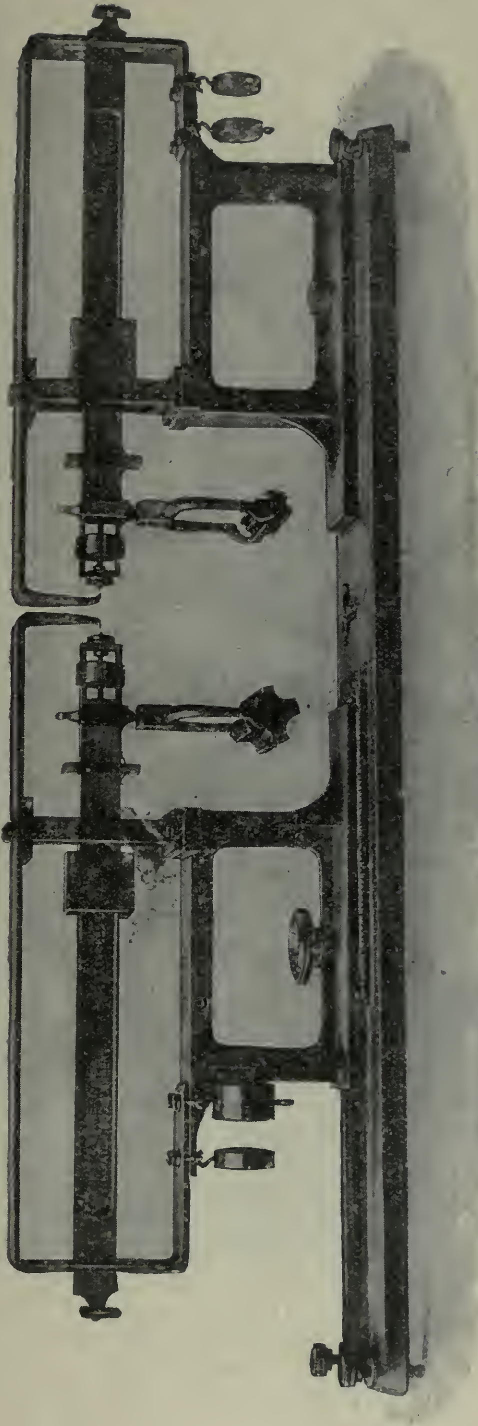


Fig. 38. Connecting Rod Balancing Machine.

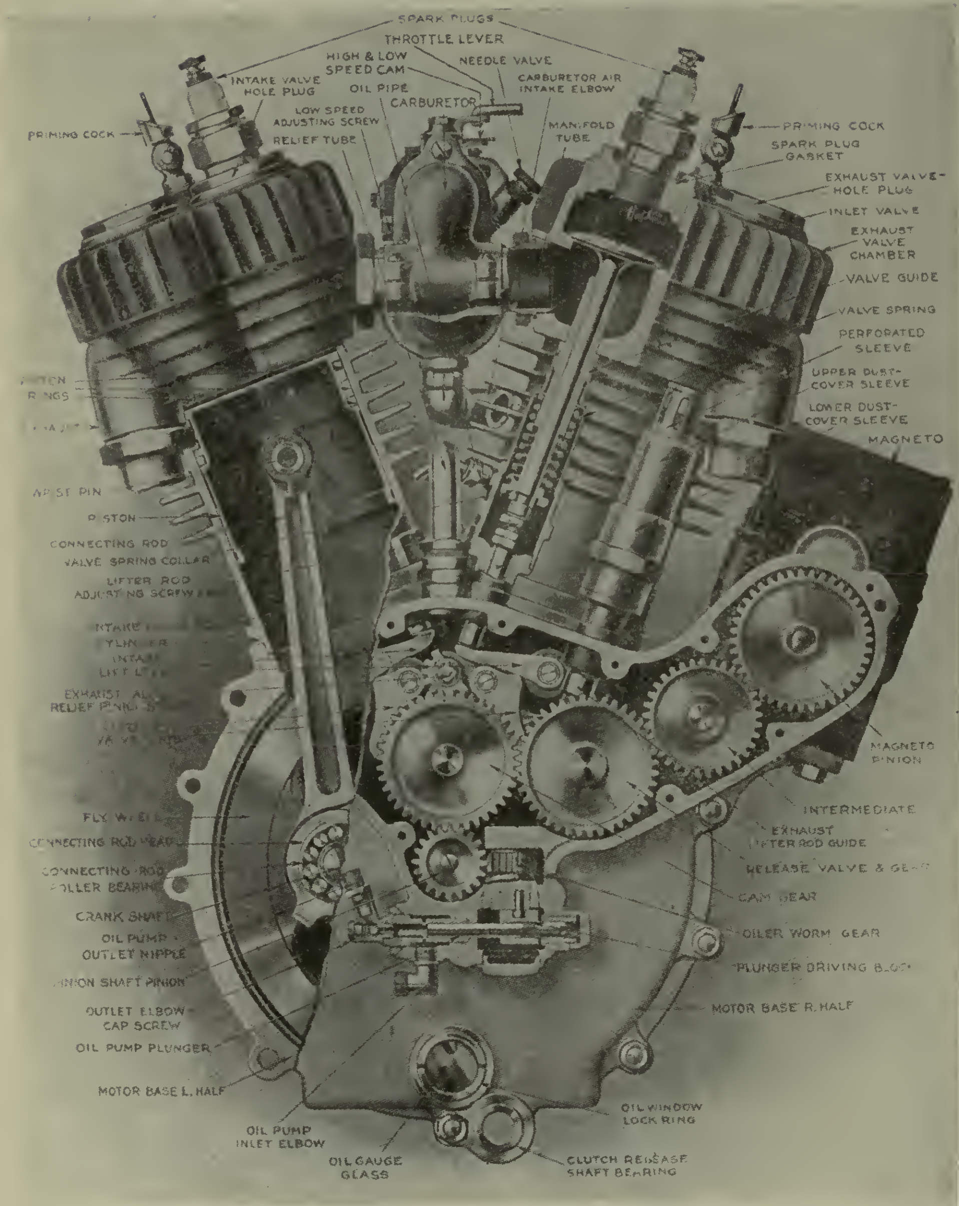


Fig. 39. Motor Cycle Engine.

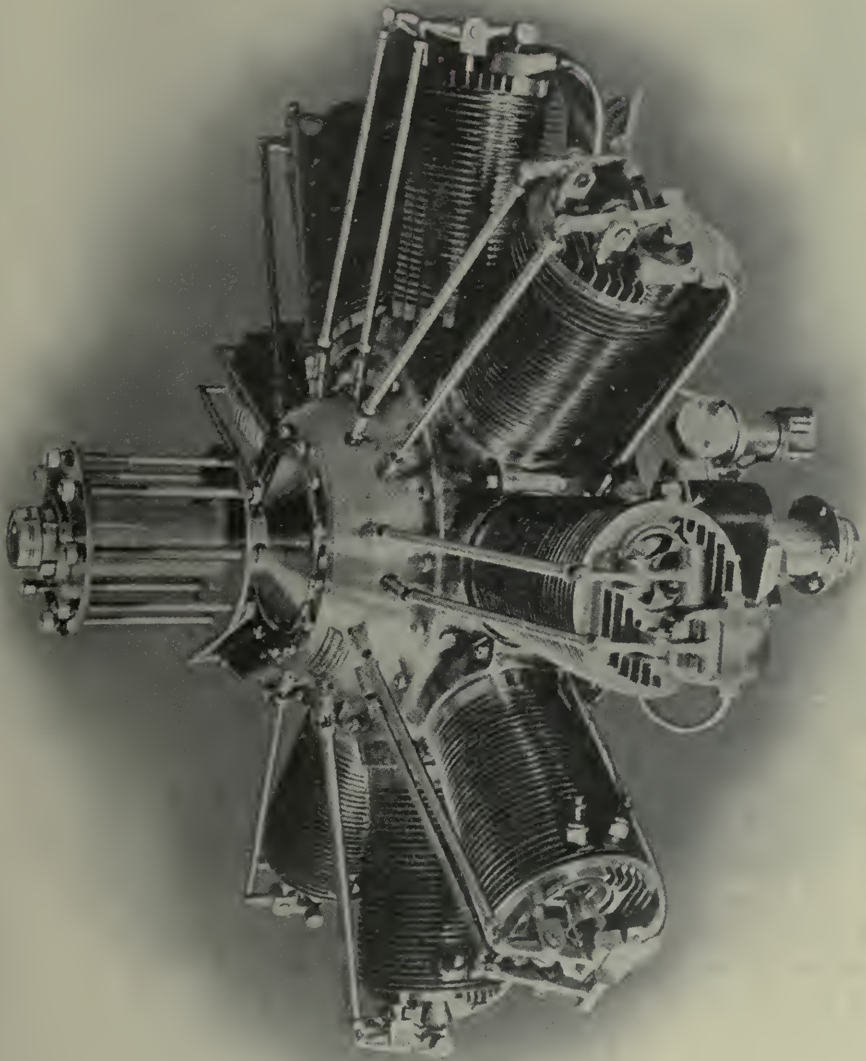


Fig. 40. Aero Engine Rotary Type, 140 B.H.P.

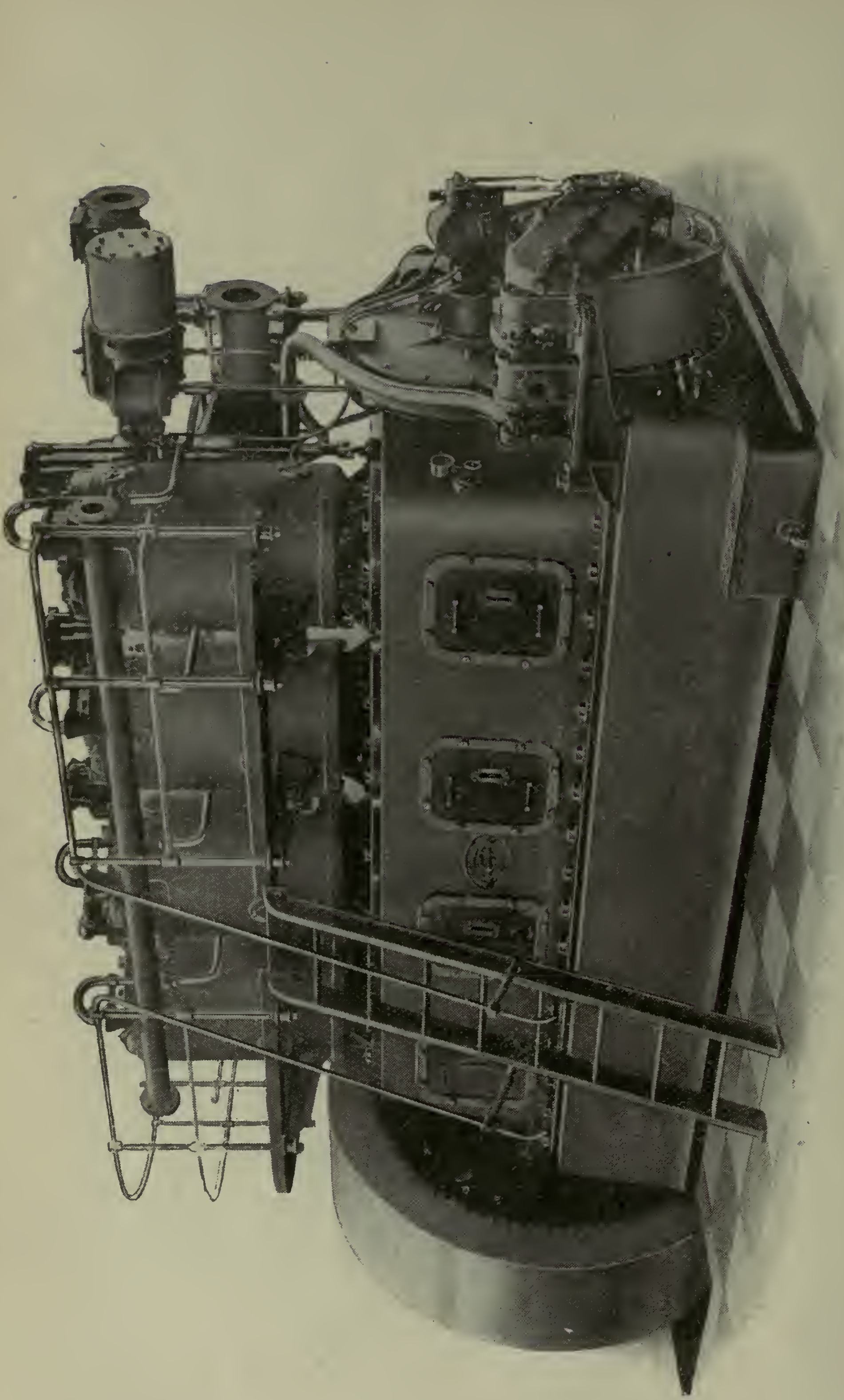


Fig. 41. Four Cylinder Gas Engine.

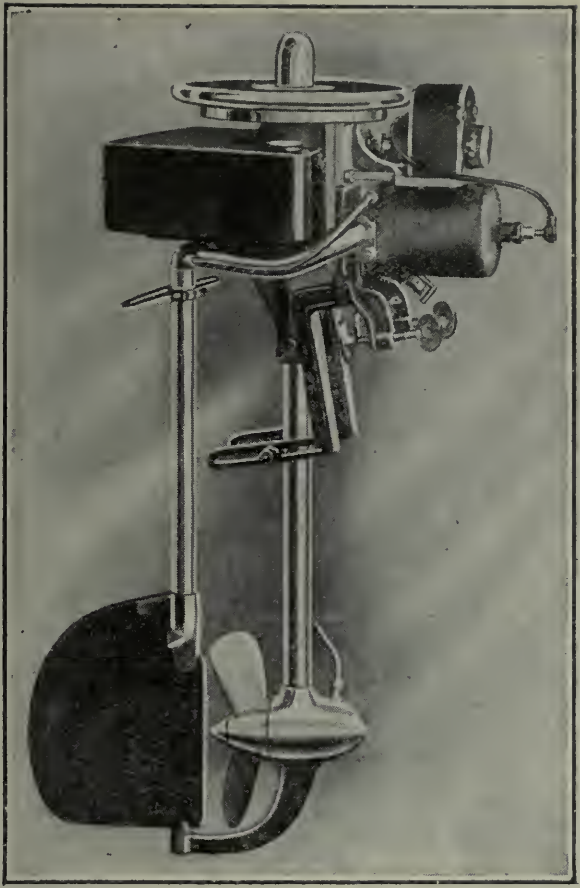


Fig. 42 Detachable Boat Motor.

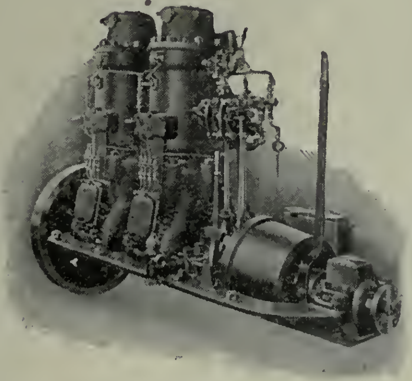


Fig. 43. Two Cylinder Hot Bulb Engine,

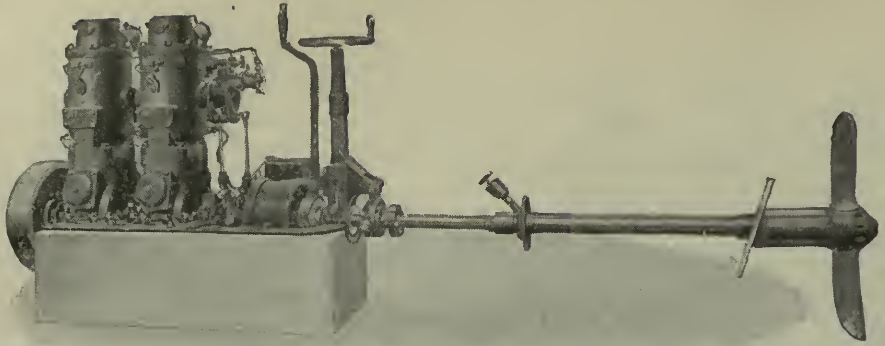


Fig. 46. Two Cylinder Hot Bulb Marine Oil Engine.

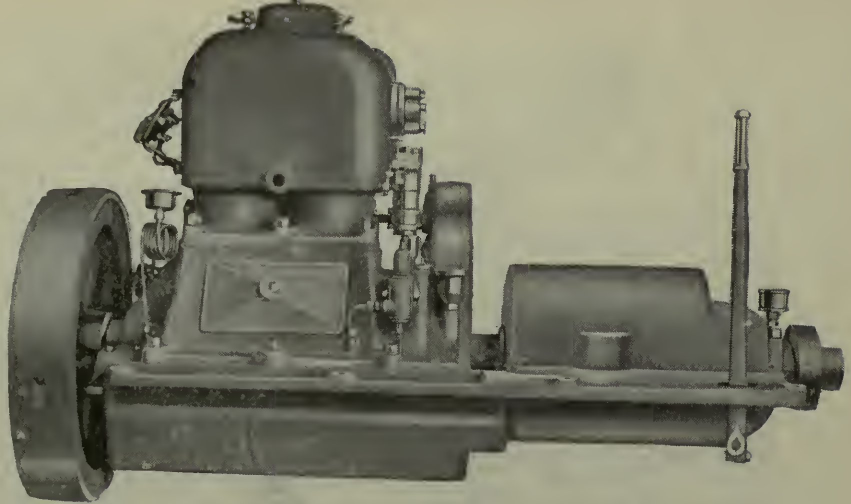


Fig. 44. Two Cylinder 15 H.P Launch Engine. Bore 5·5 in., Stroke 7 ins.

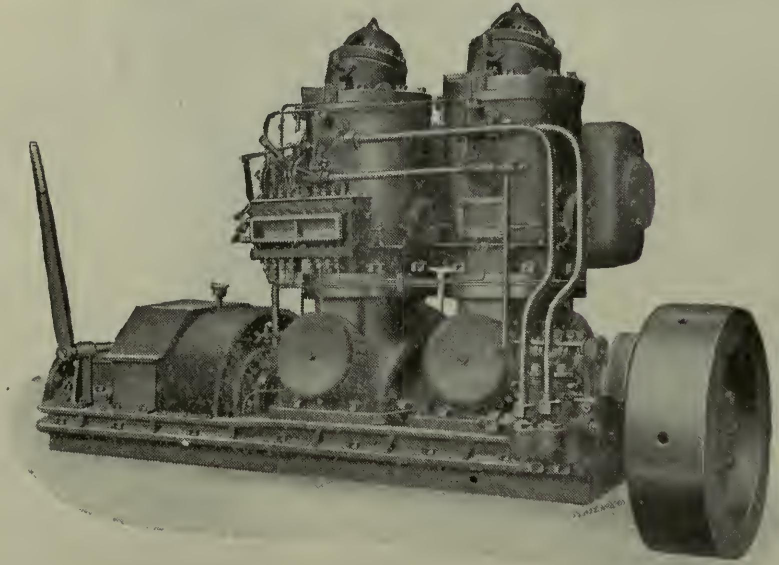


Fig 47. Two Cylinder Direct Reversing Marine Oil Engine

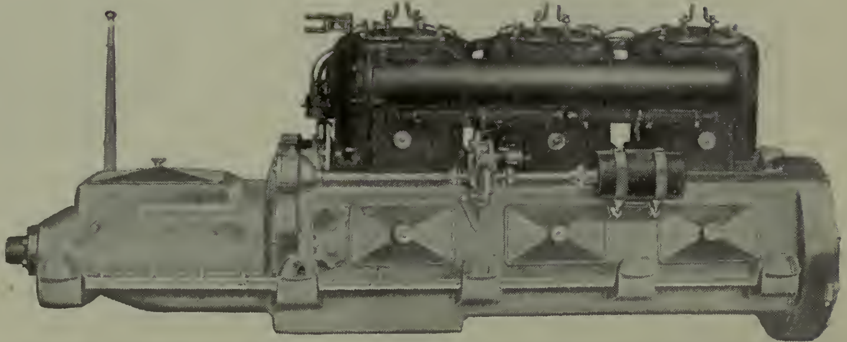


Fig. 45. Six Cylinder 85 B.H.P. Launch Engine. Bore, 5·5 in., Stroke, 3·75 ins.

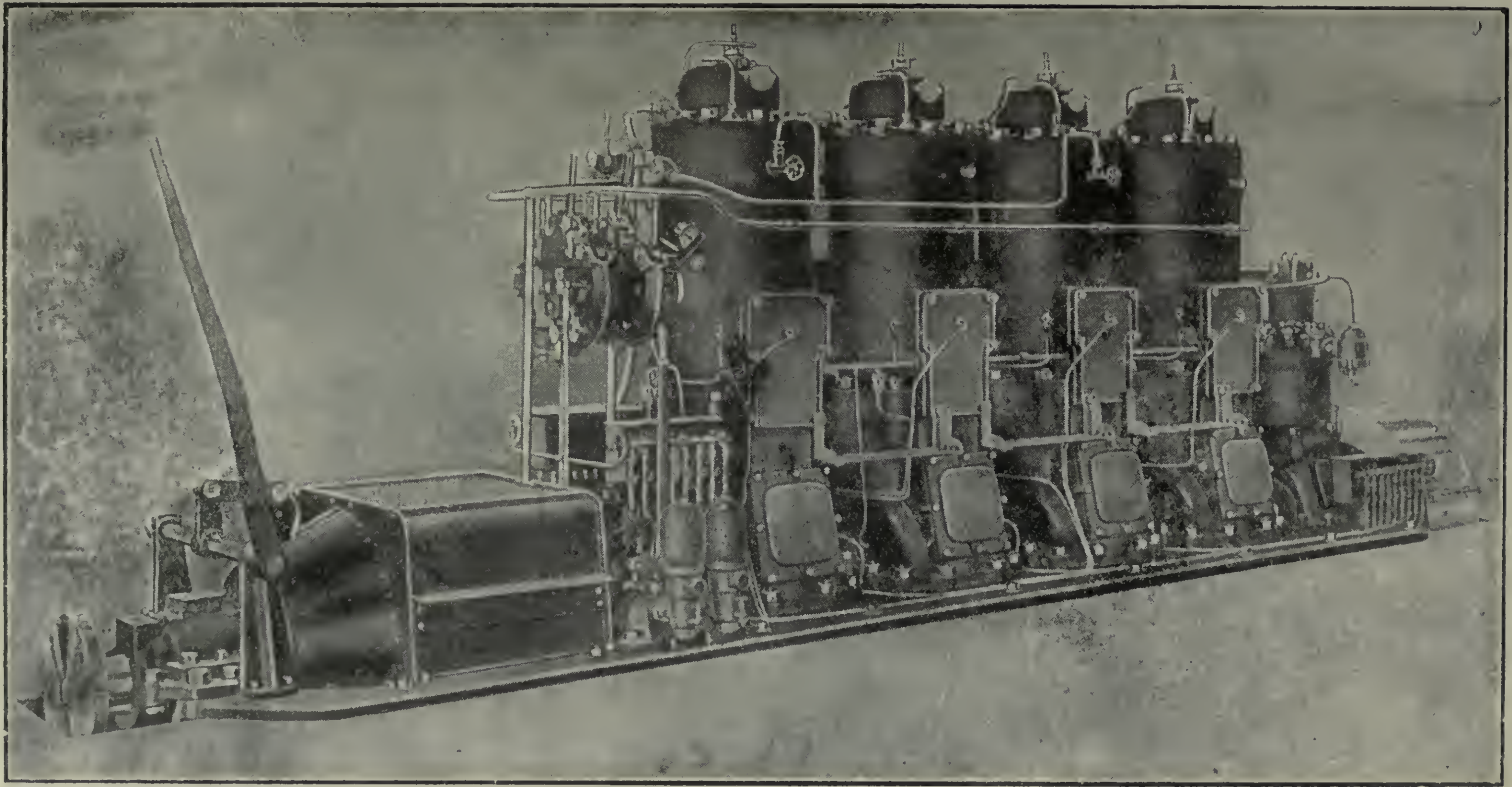


Fig. 48. Four Cylinder Marine Oil Engine.

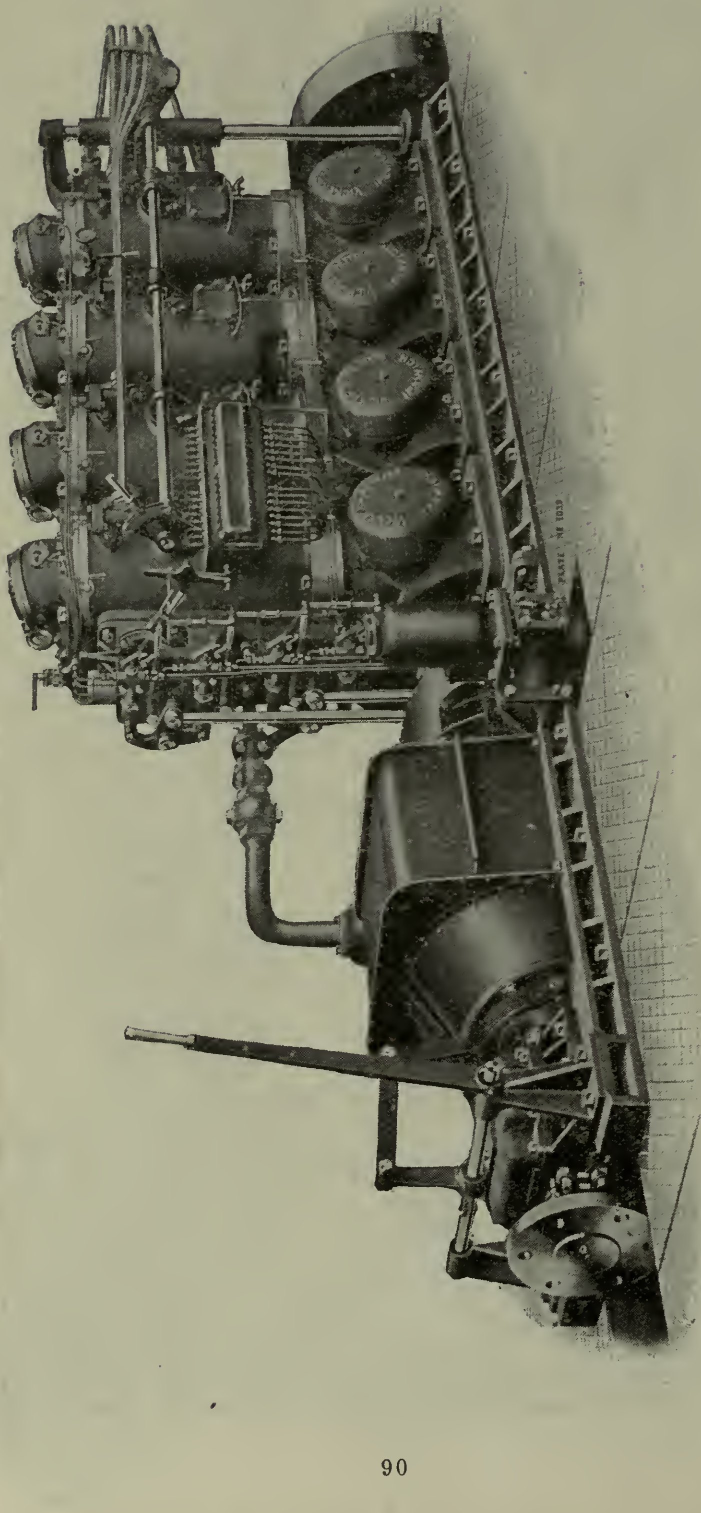


Fig. 49. 220 B,H.P. Direct Reversing Marine Crude Oil Engine.

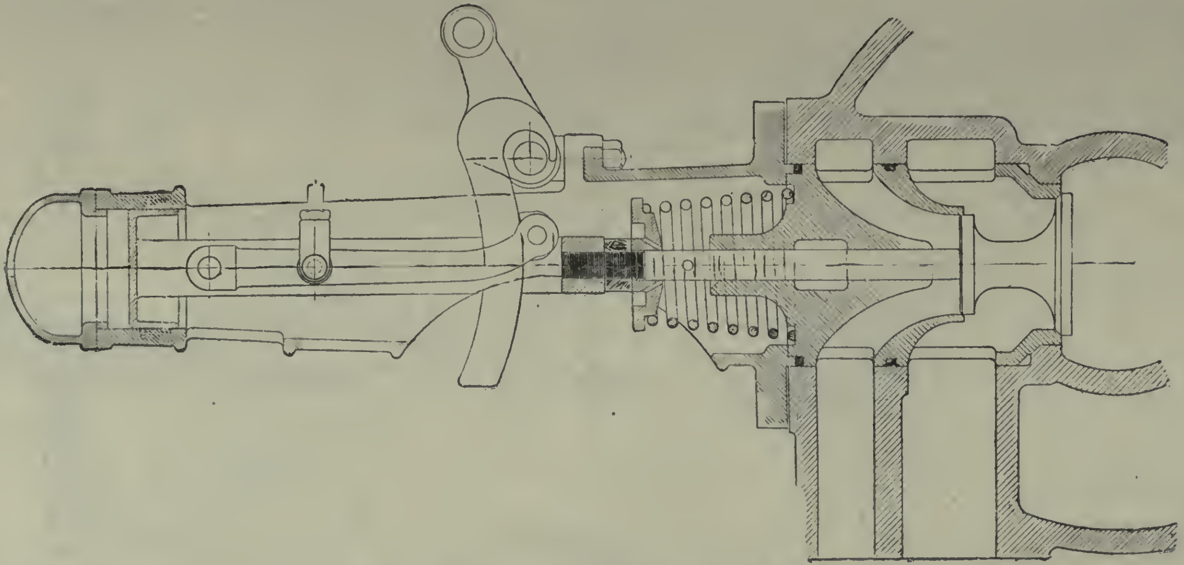


Fig. 50. Gas Engine Inlet Valve. Quantity Governing System.

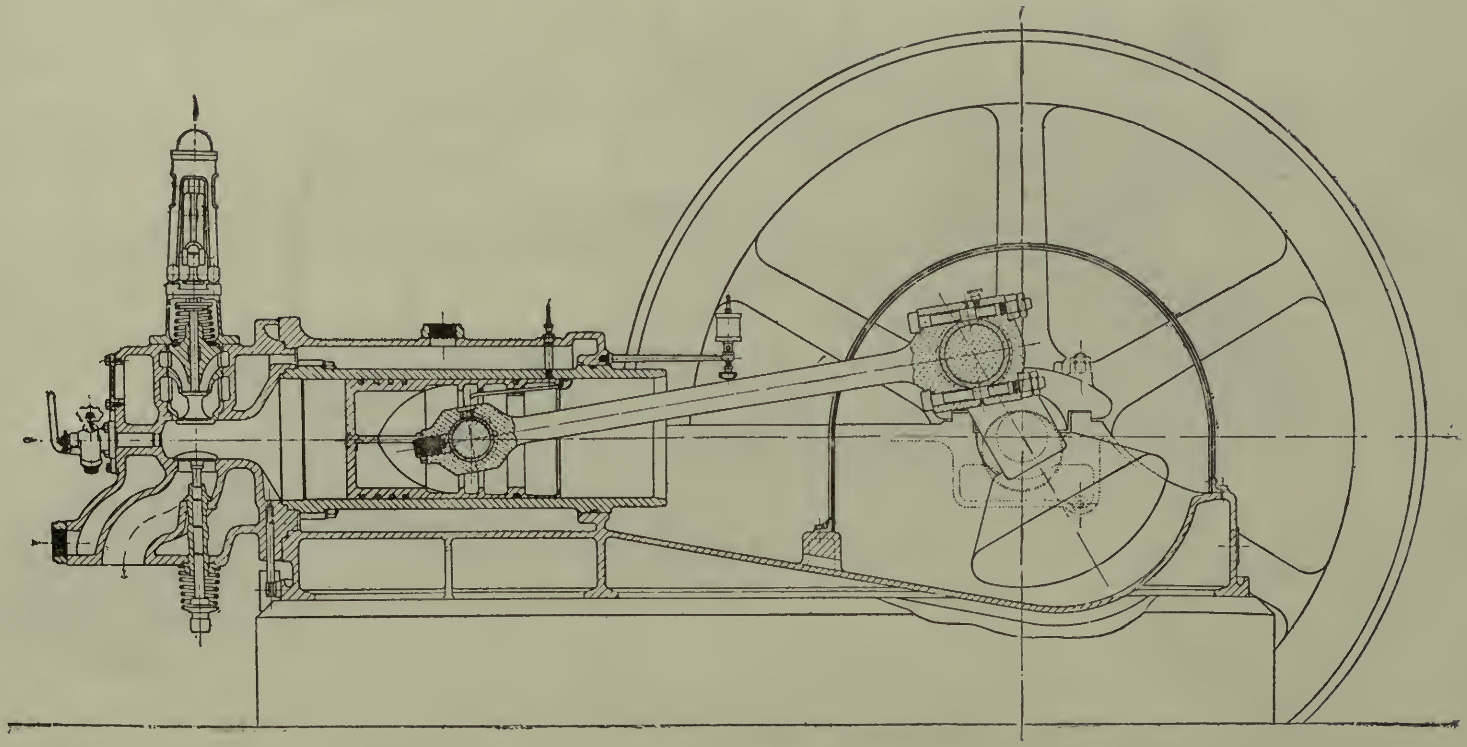


Fig. 51. Section through Gas Engine.

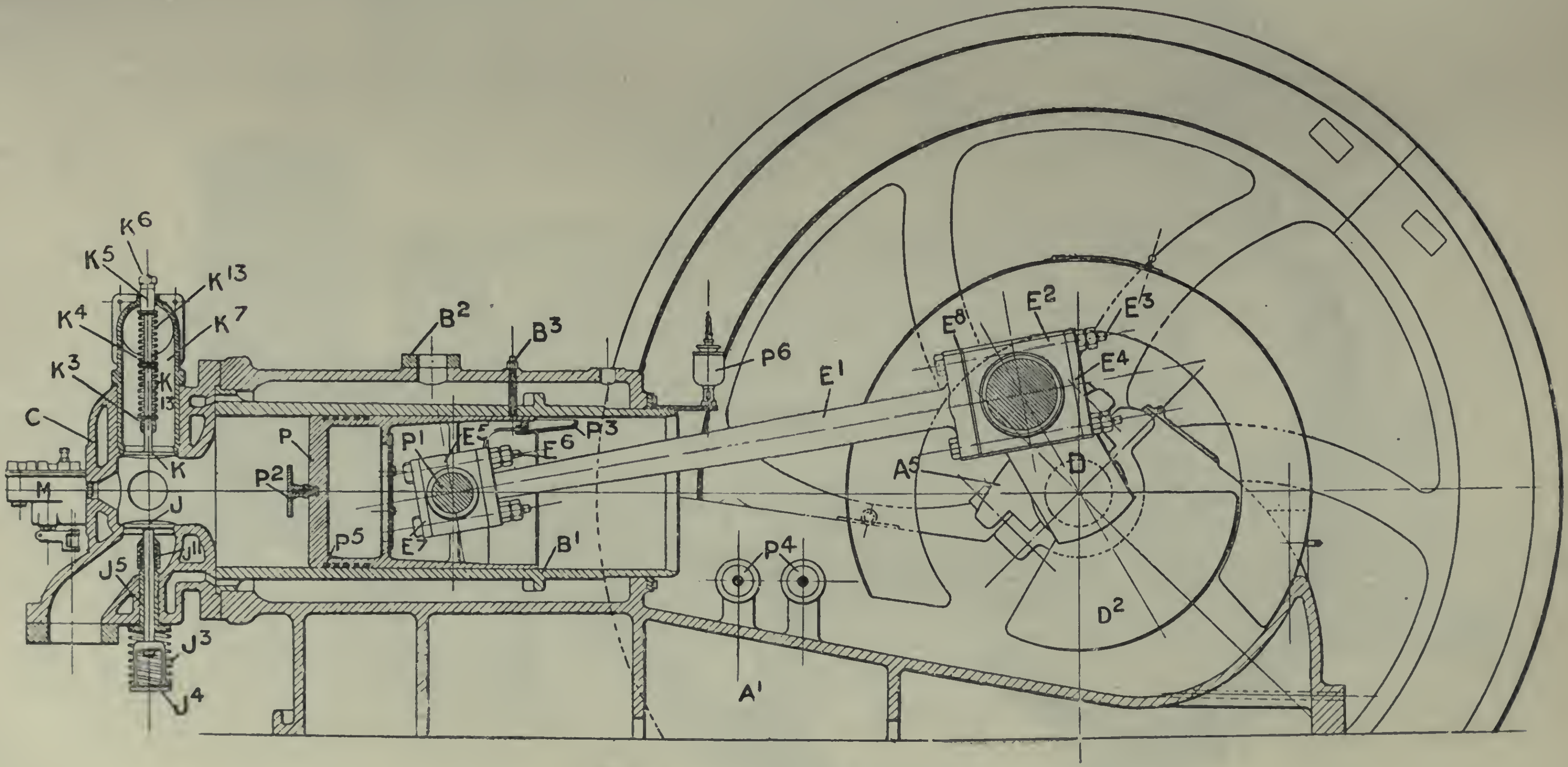


Fig. 52. Section through Gas Engine.

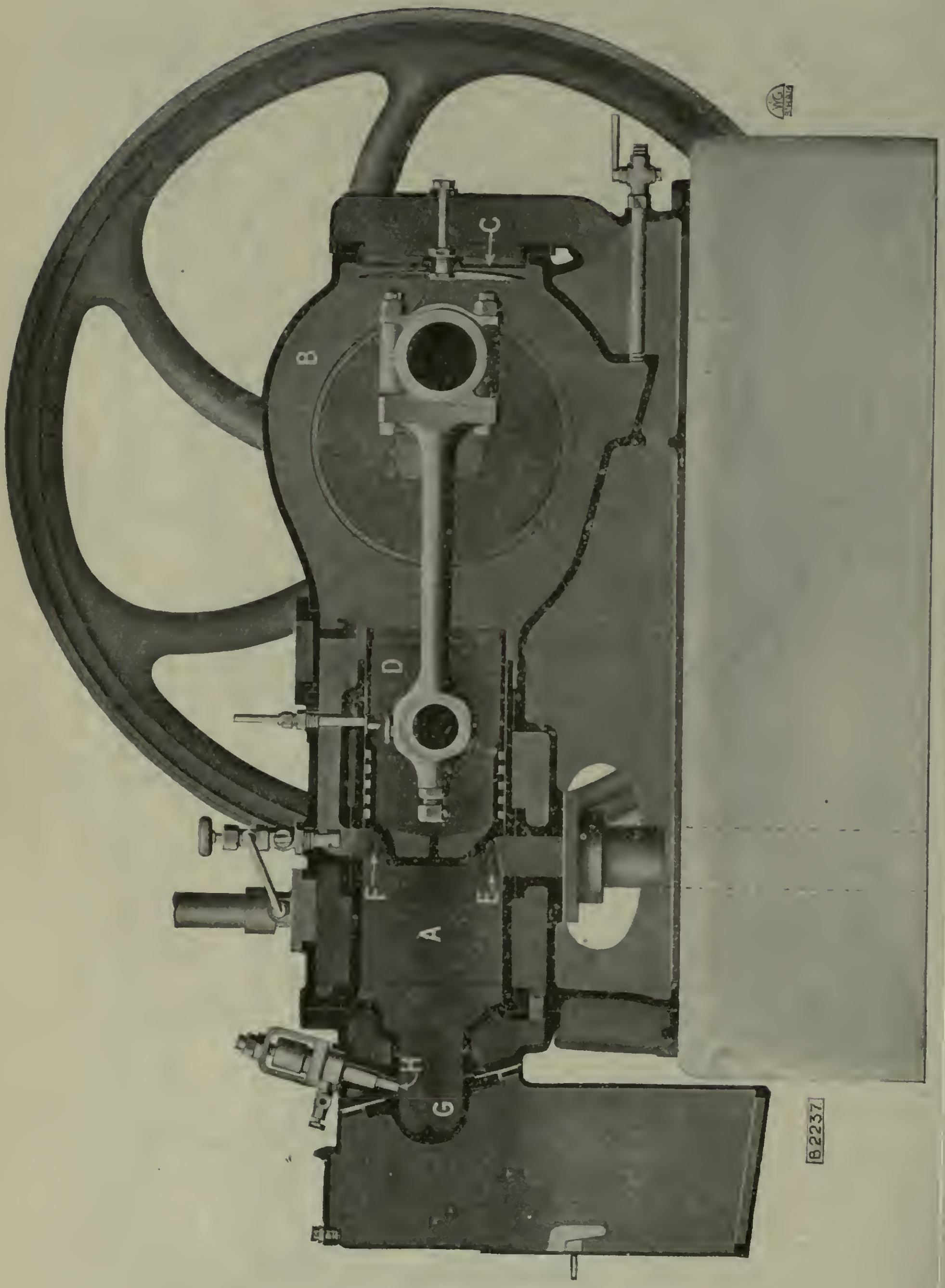


Fig. 53. Section through Two Cycle Oil Engine.

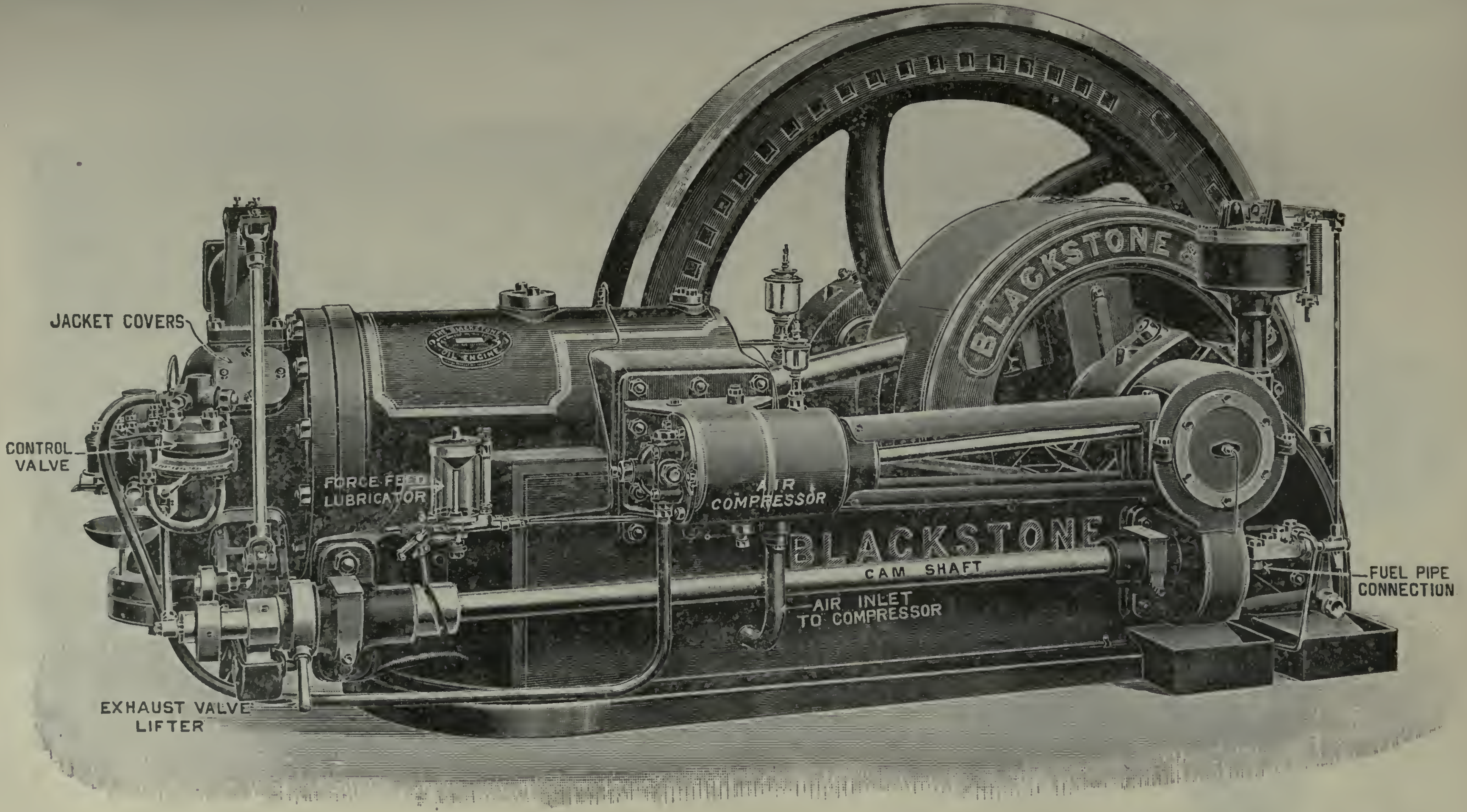


Fig. 54. Oil Engine.[6]

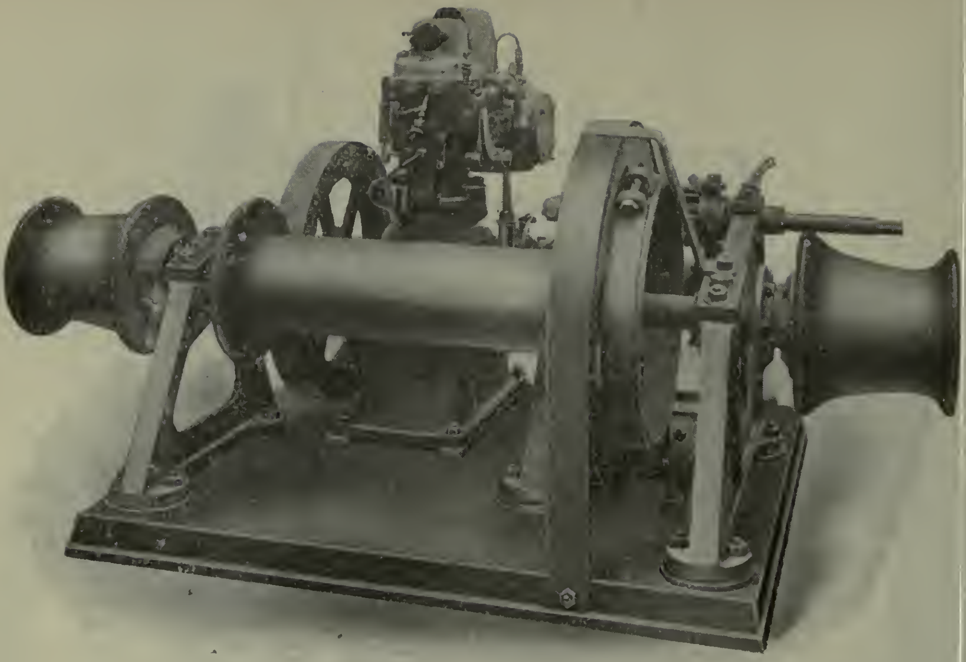


Fig. 55. Motor Winch.

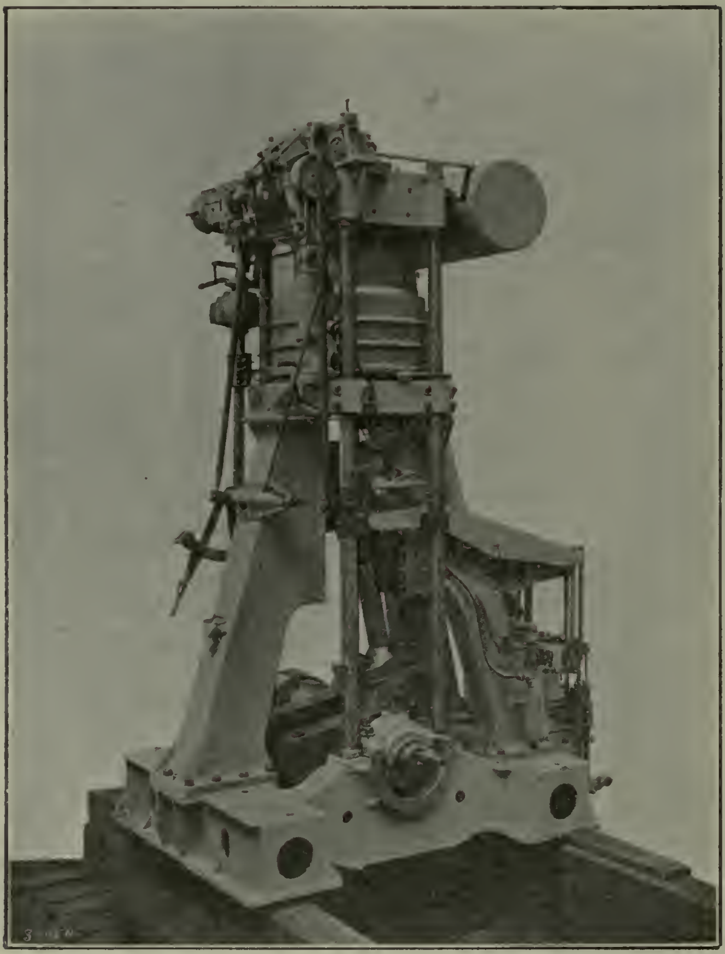


Fig. 56. Experimental Two Stroke Marine Oil Engine (Diesel).

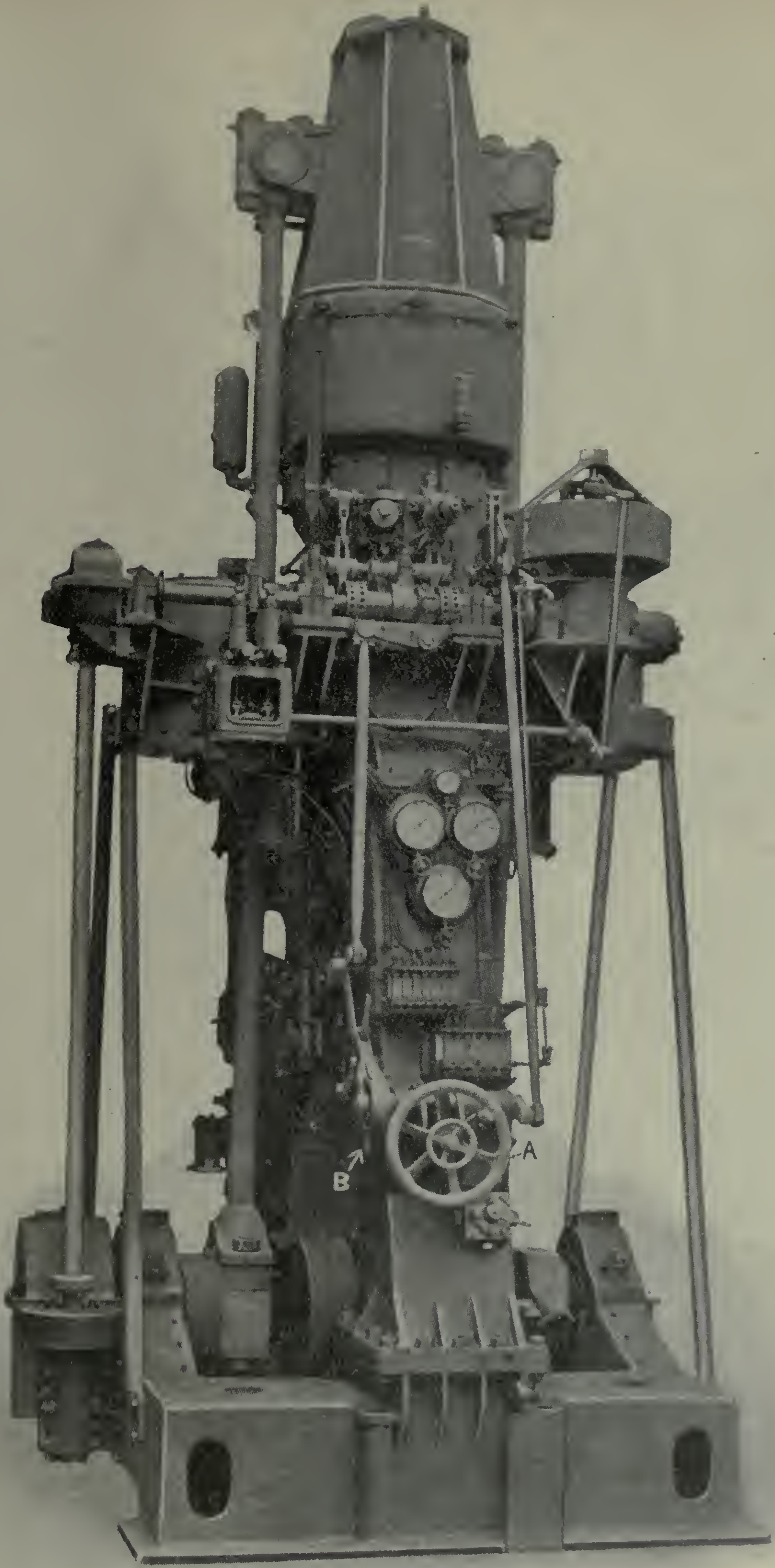


Fig. 57. Experimental Two Stroke Marine Oil Engine, Opposed Piston Type (Diesel).

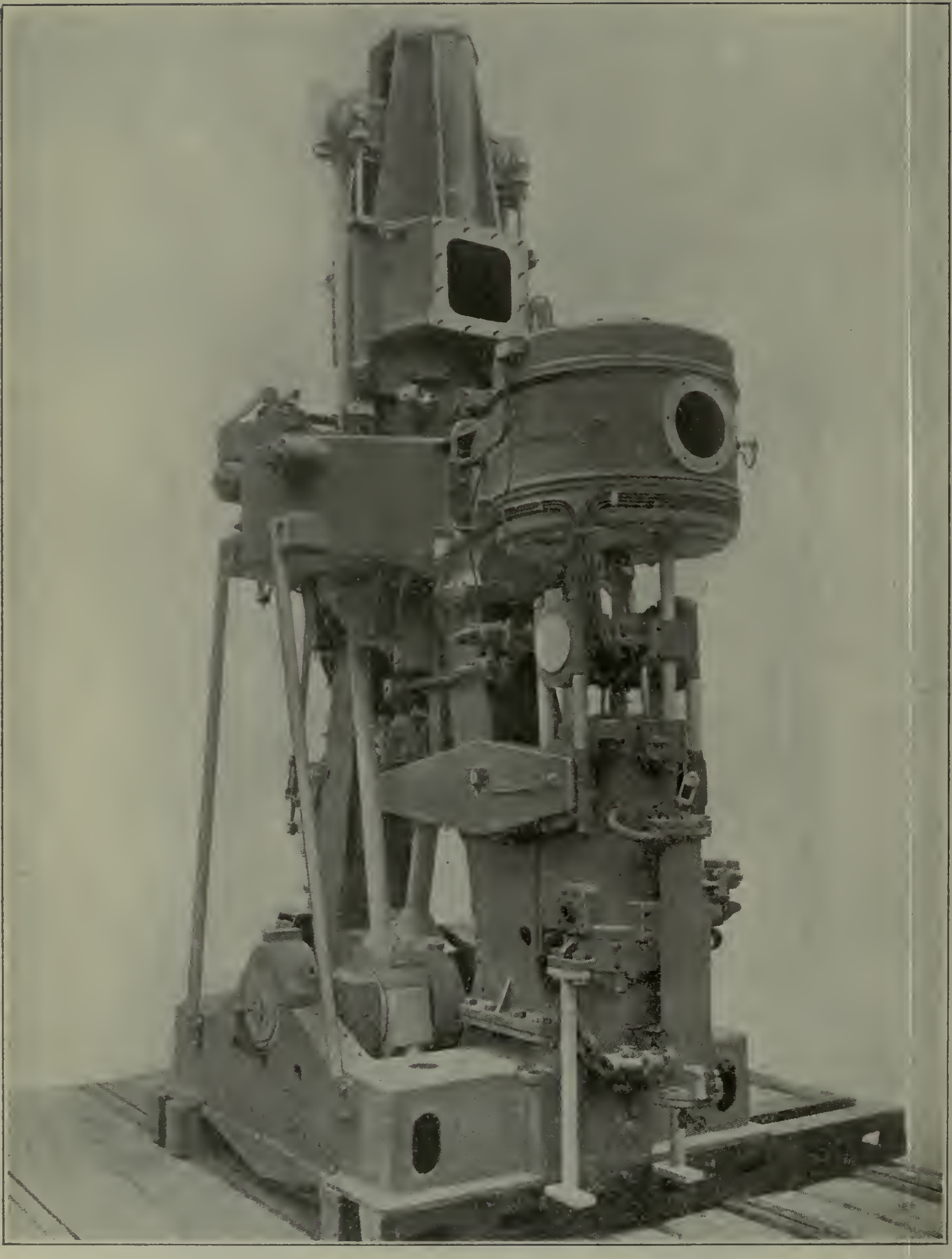


Fig. 58. Back View.

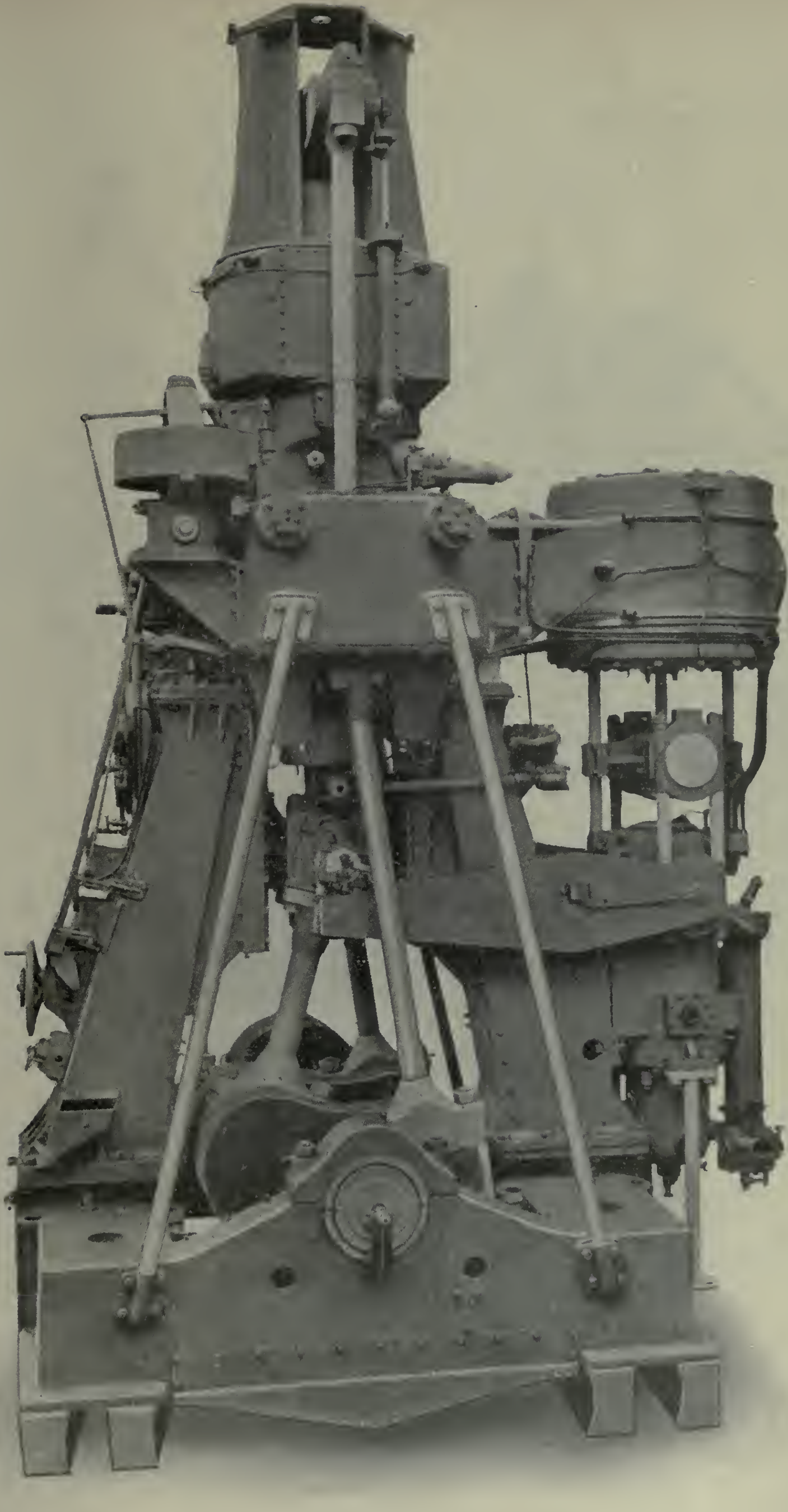


Fig. 59. Side View.

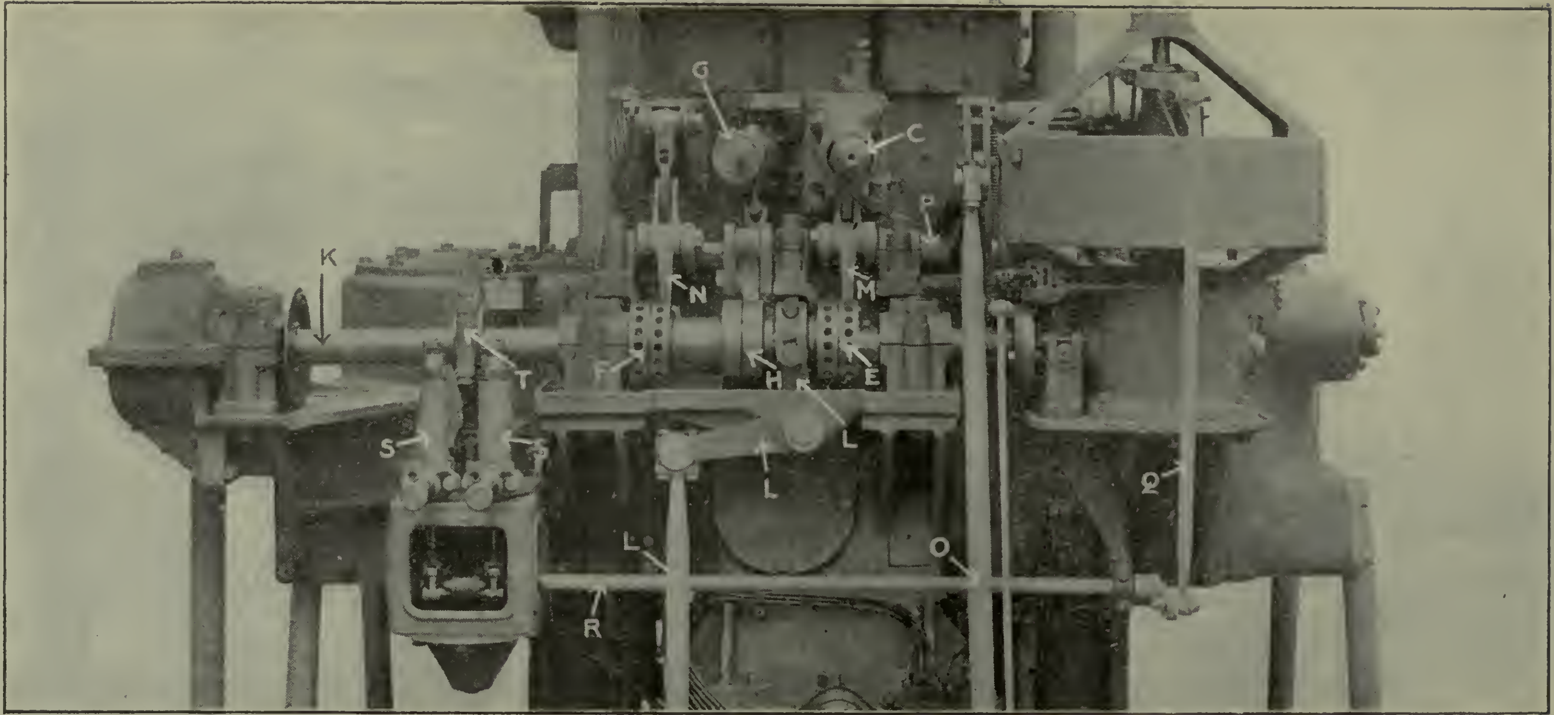


Fig. 60. Valve Gear.

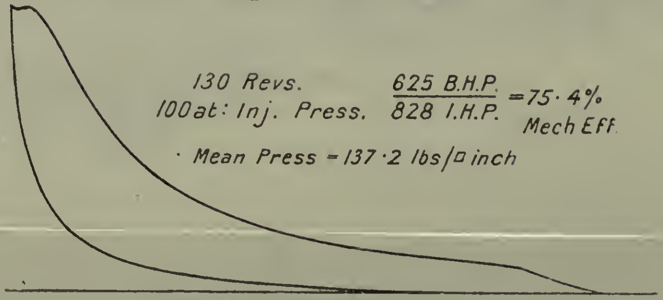


Fig. 61. Indicator Diagram

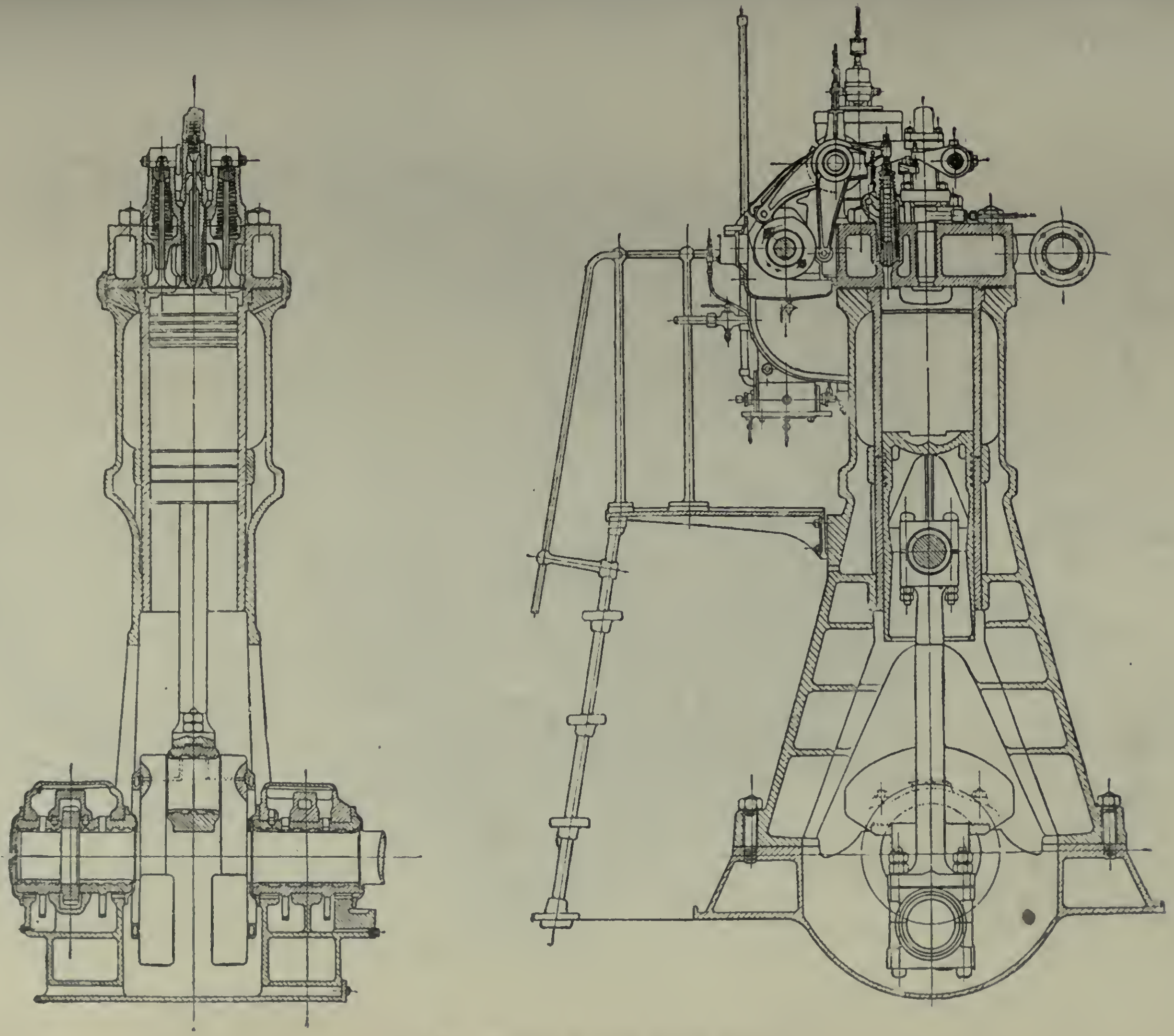


Fig. 62. Section through Diesel Engine.

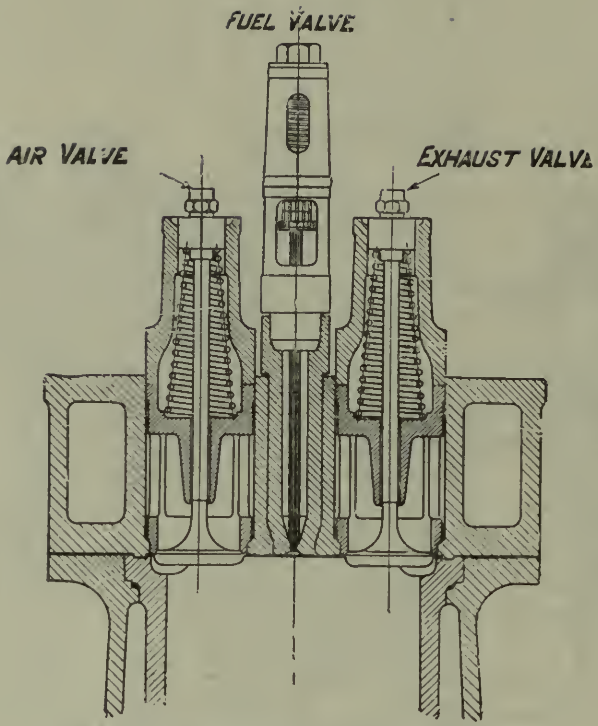


Fig. 63. Section through Valves, Diesel Engine.

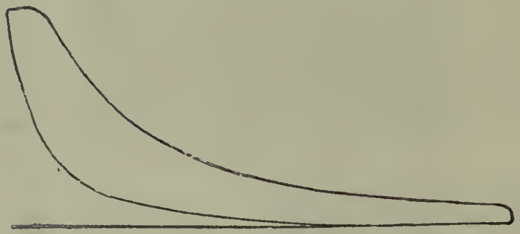


Fig. 64. Diesel Full Load Diagram

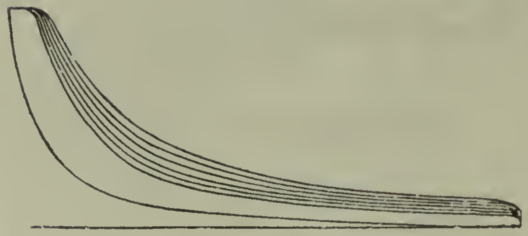


Fig. 65. Diesel Diagram, showing Governing Action.

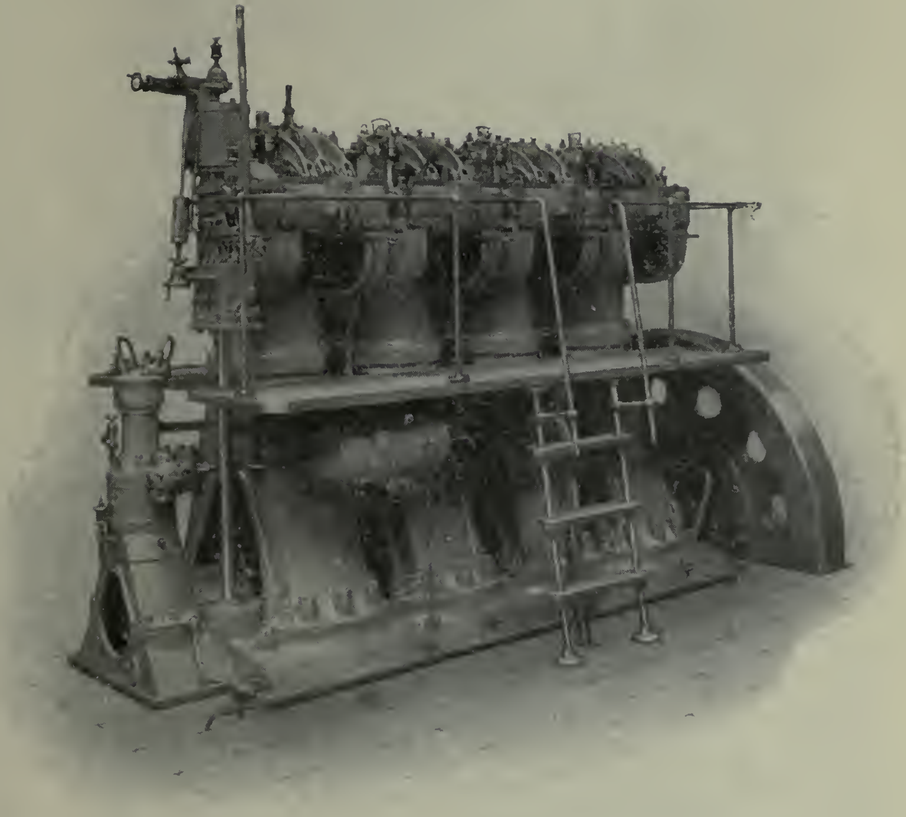


Fig. 66. 200 B.H.P. Diesel Engine.

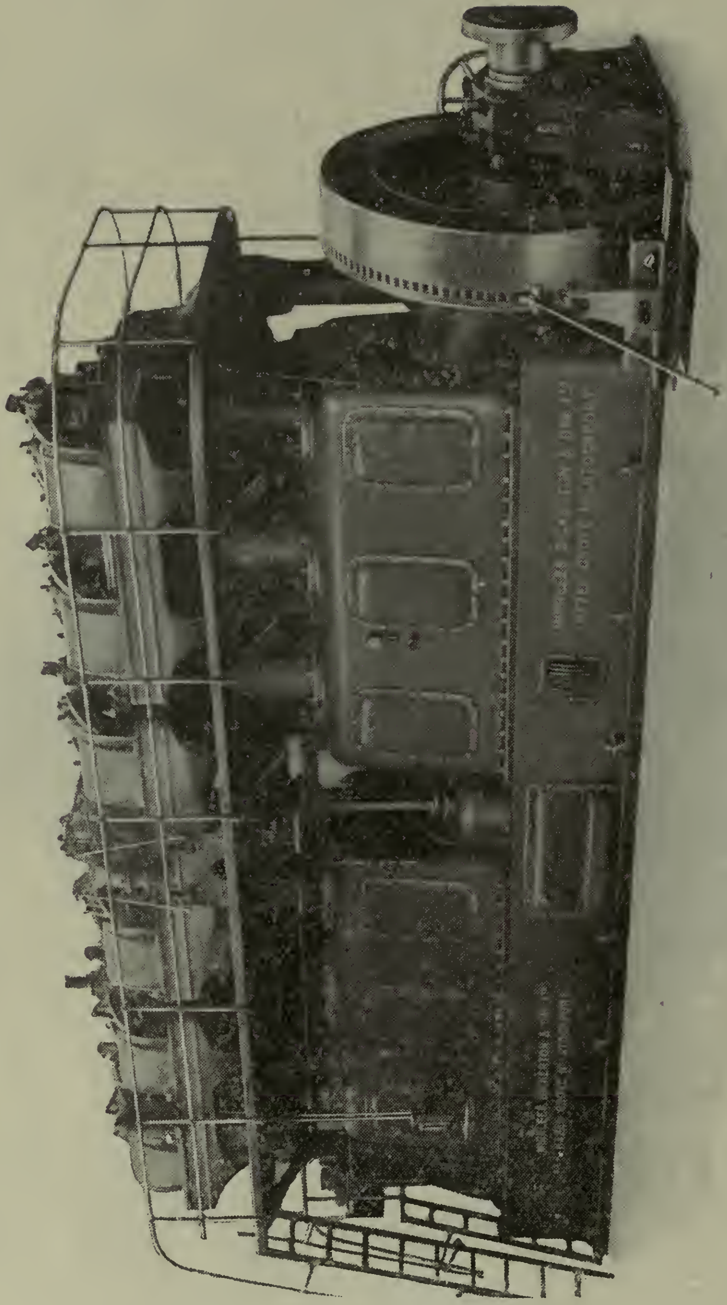


Fig. 67. 750 B.H.P. Diesel Engine.

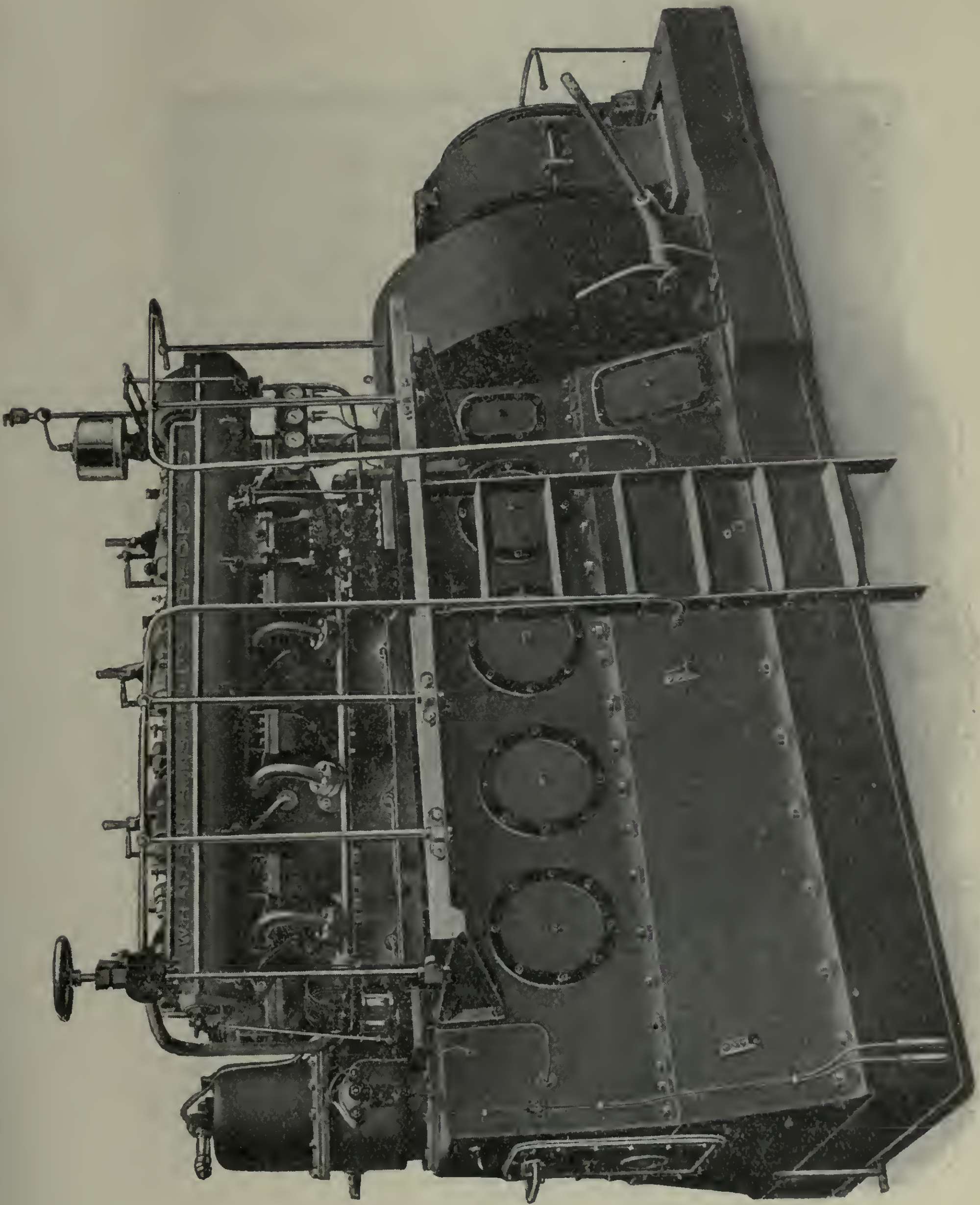


Fig. 68 Auxiliary Diesel Engine.

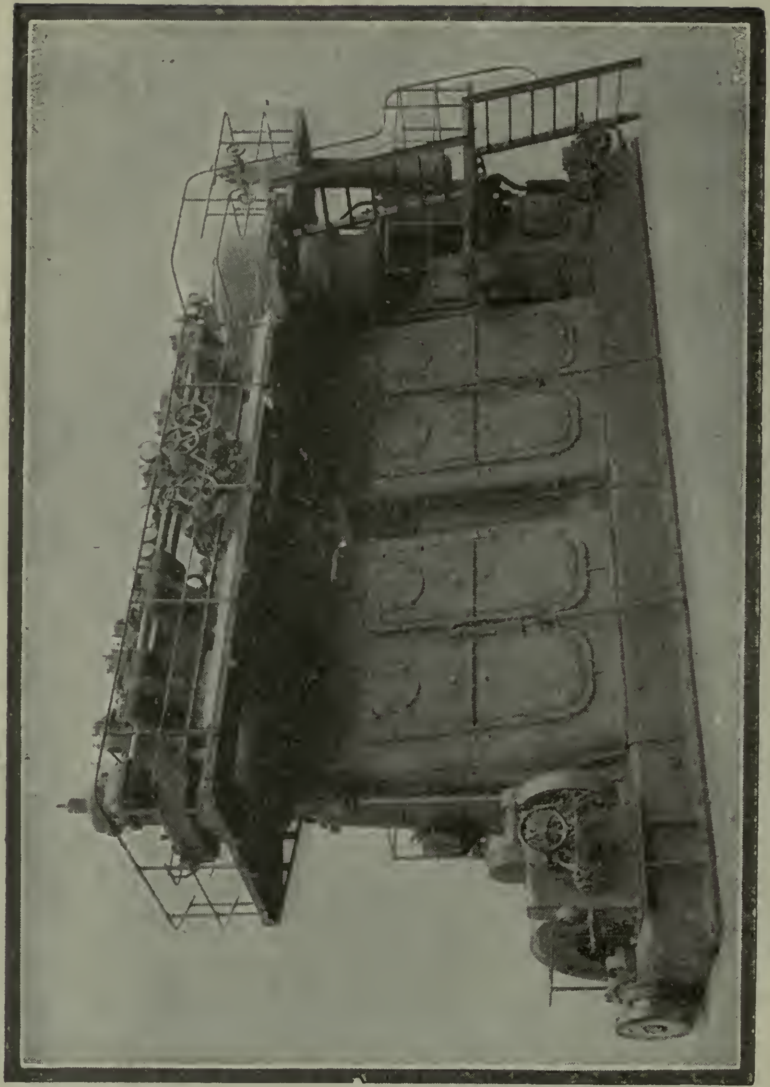


Fig. 69. Two-cycle Diesel Engine, 1,250 h p., 100 r.p.m.

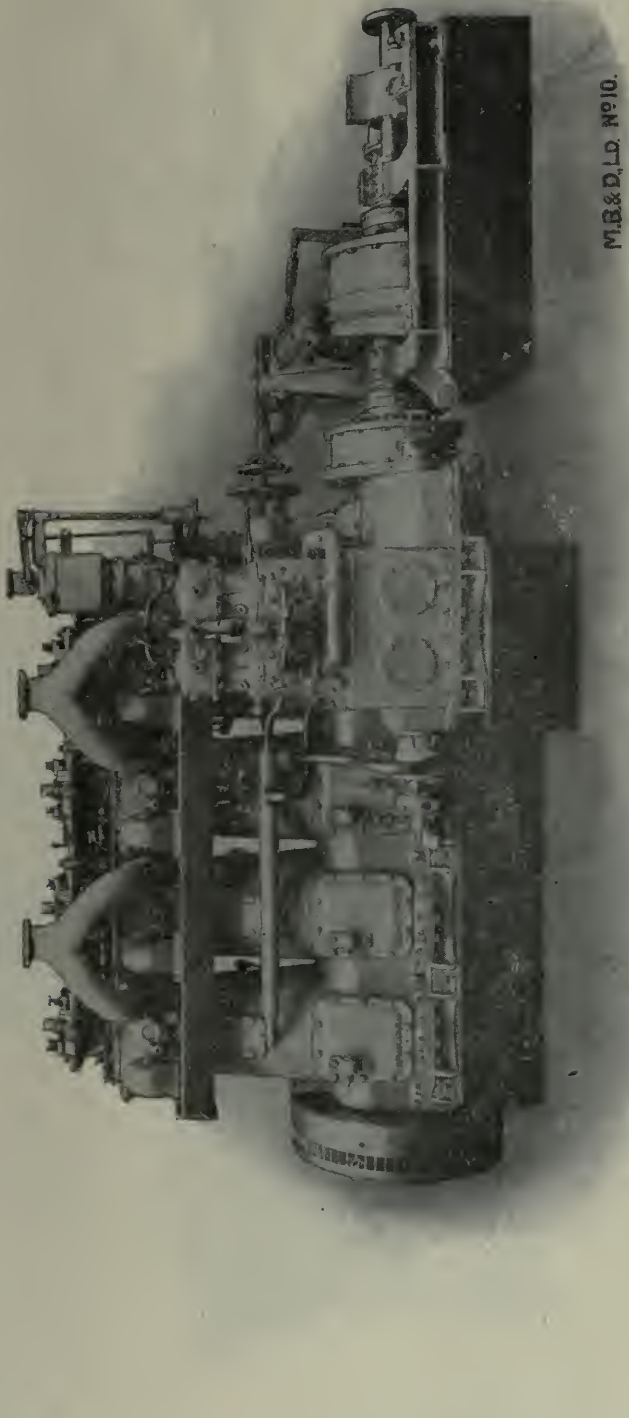


Fig. 70. 120 B.H.P. 50 ft. Pinnace Diesel Engine.

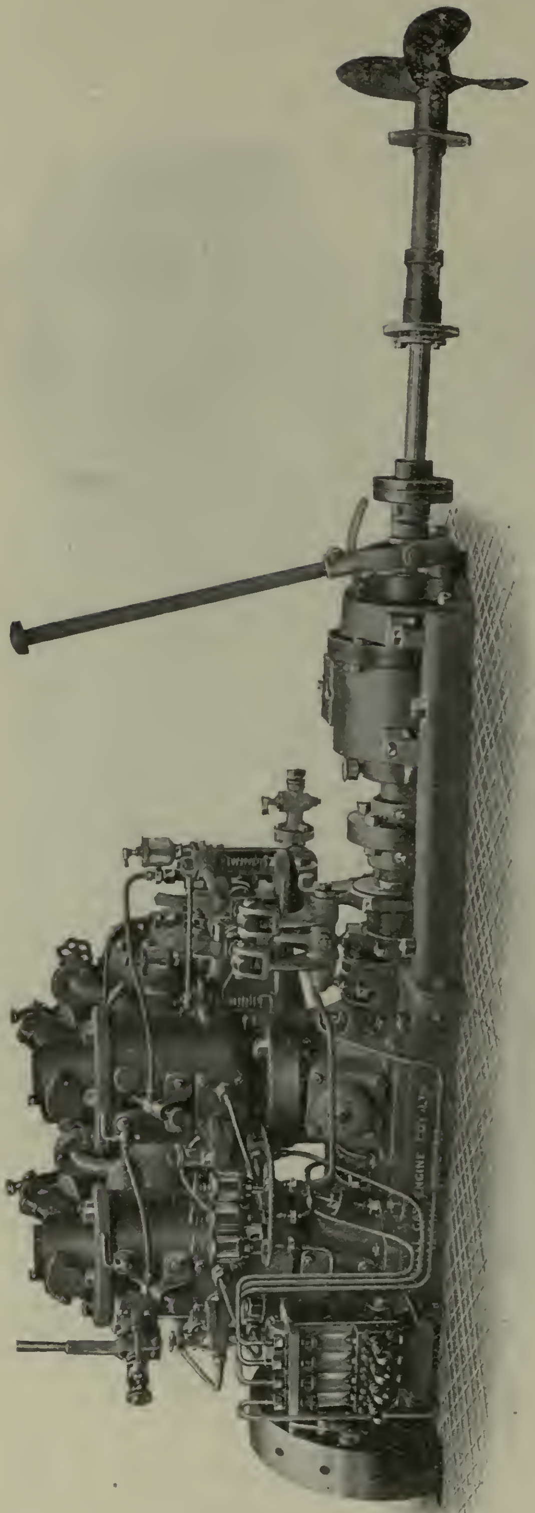


Fig. 71. 17·5 B.H.P. Two Cylinder Gear Reversing Engine.

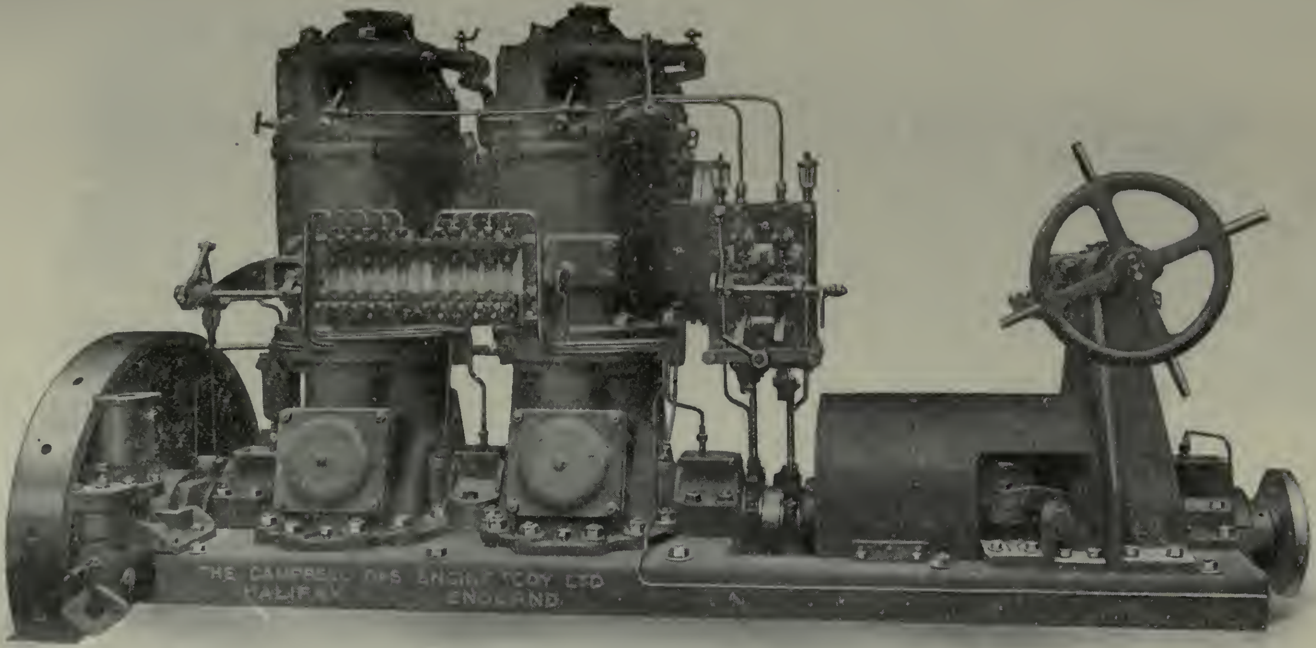


Fig. 72. 88 B.H.P. Two Cylinder Gear Reversing Engine.

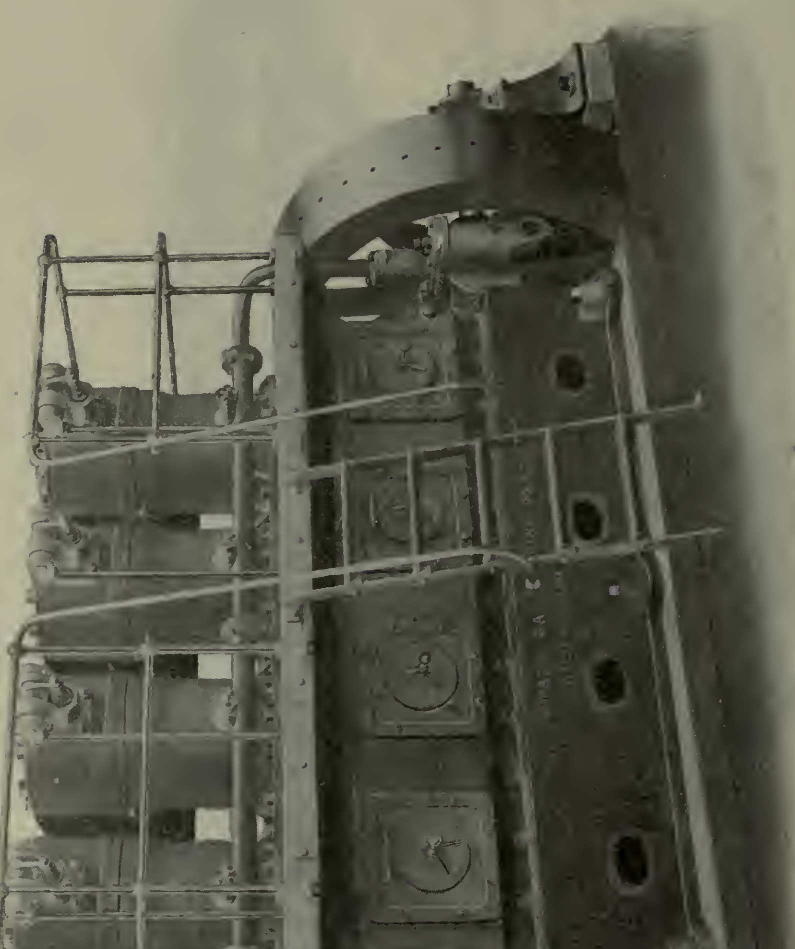


Fig. 73. 500 B.H.P. Four Cylinder Heavy Oil Engine.

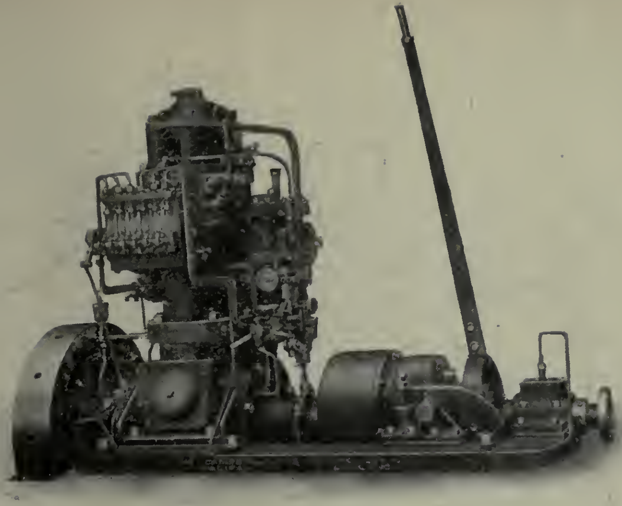


Fig. 74. Single Cylinder Hot Bulb Engine, 27·5 B.H.P.

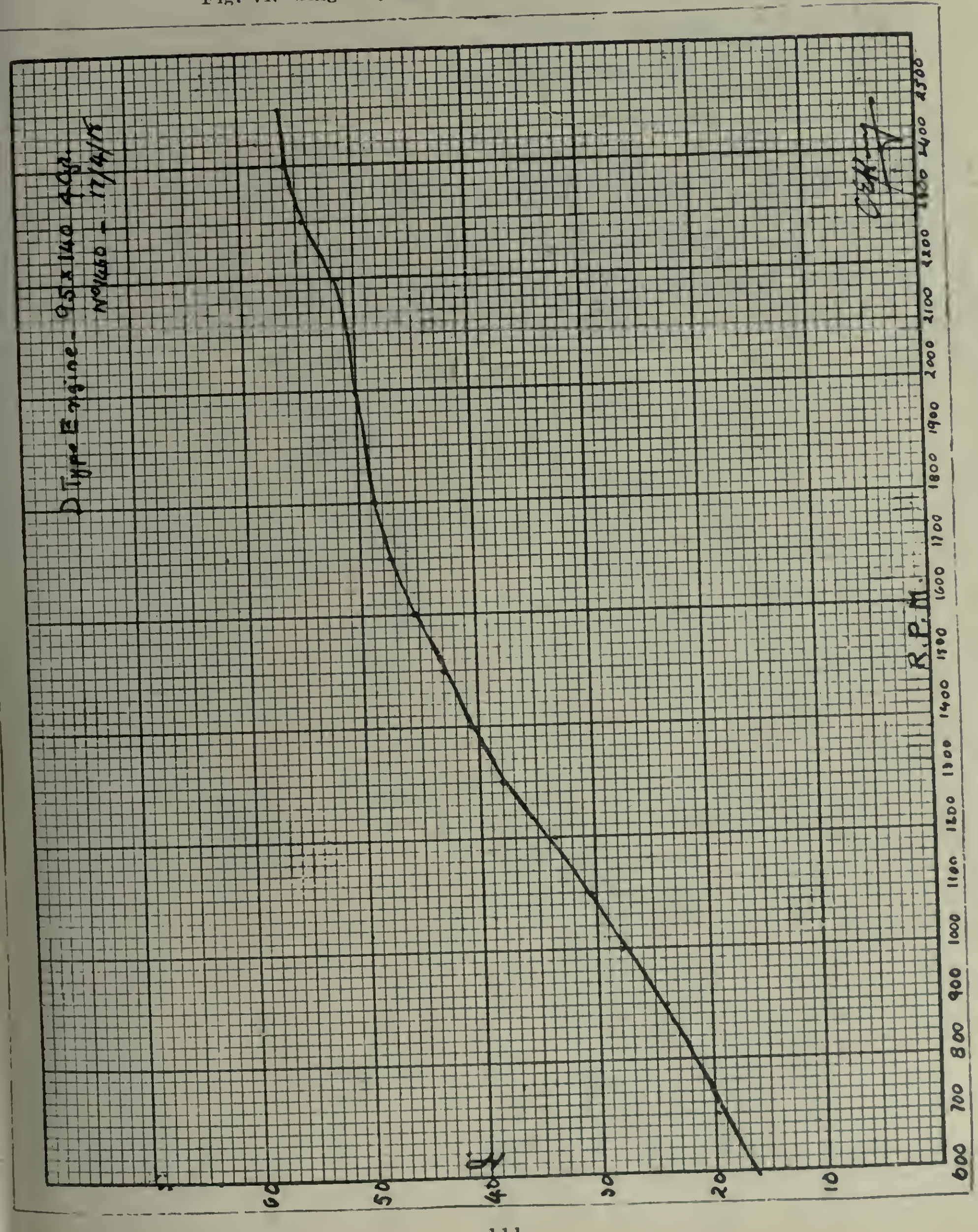


Fig. 75. Power Curve of Modern Automobile Engine.

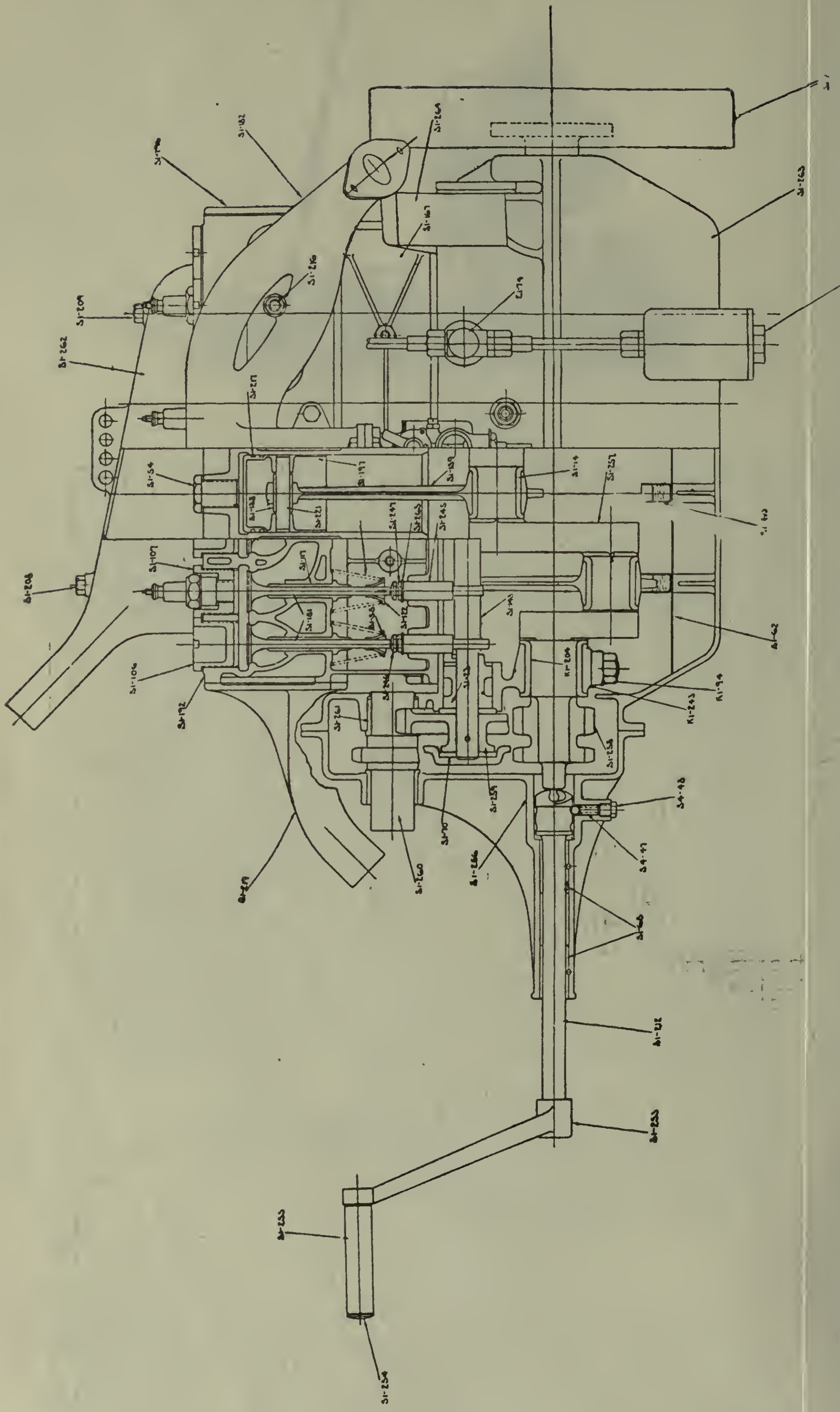


Fig 76. Side View Modern Light Car Engine.

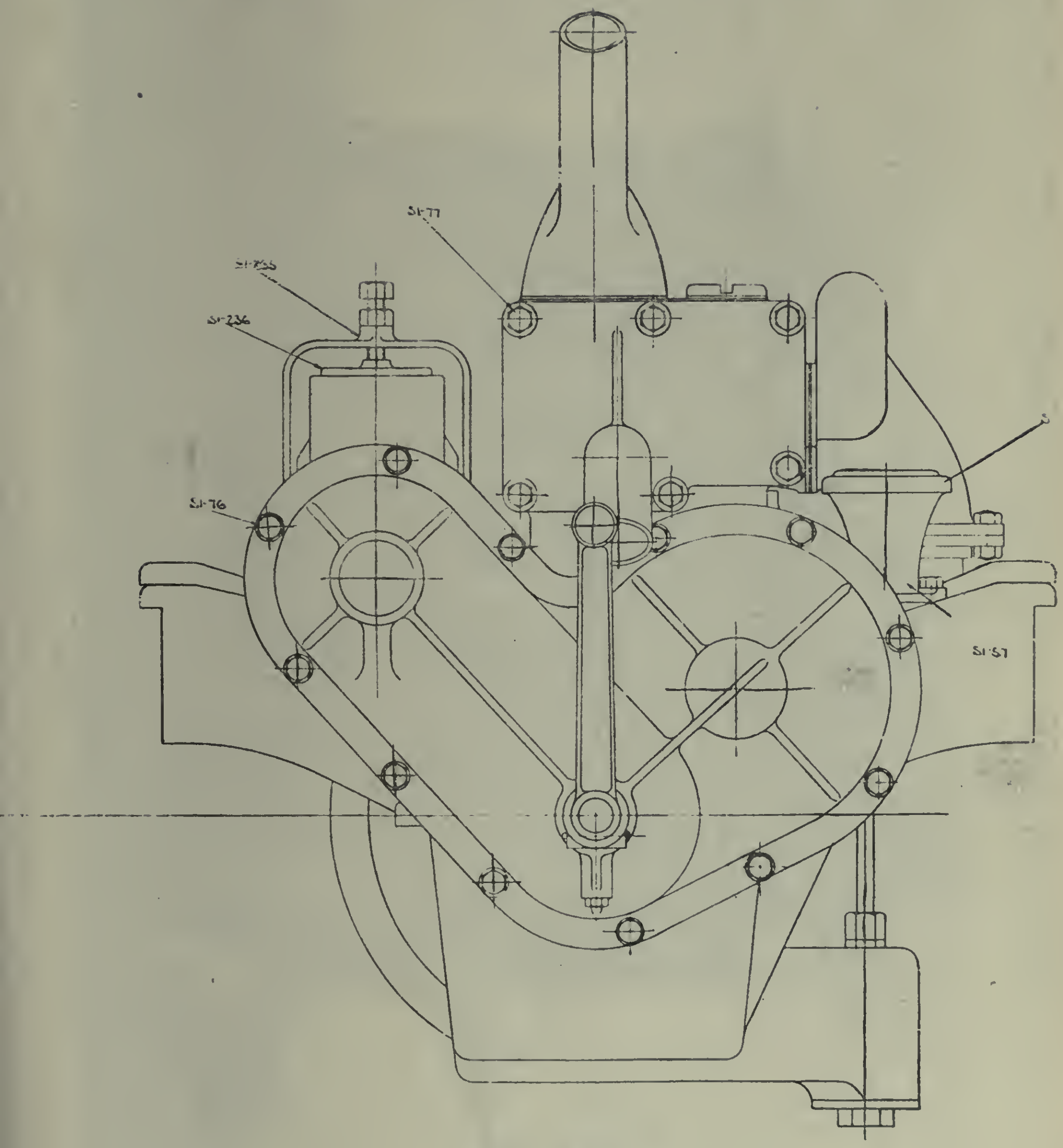


Fig. 77. End View.

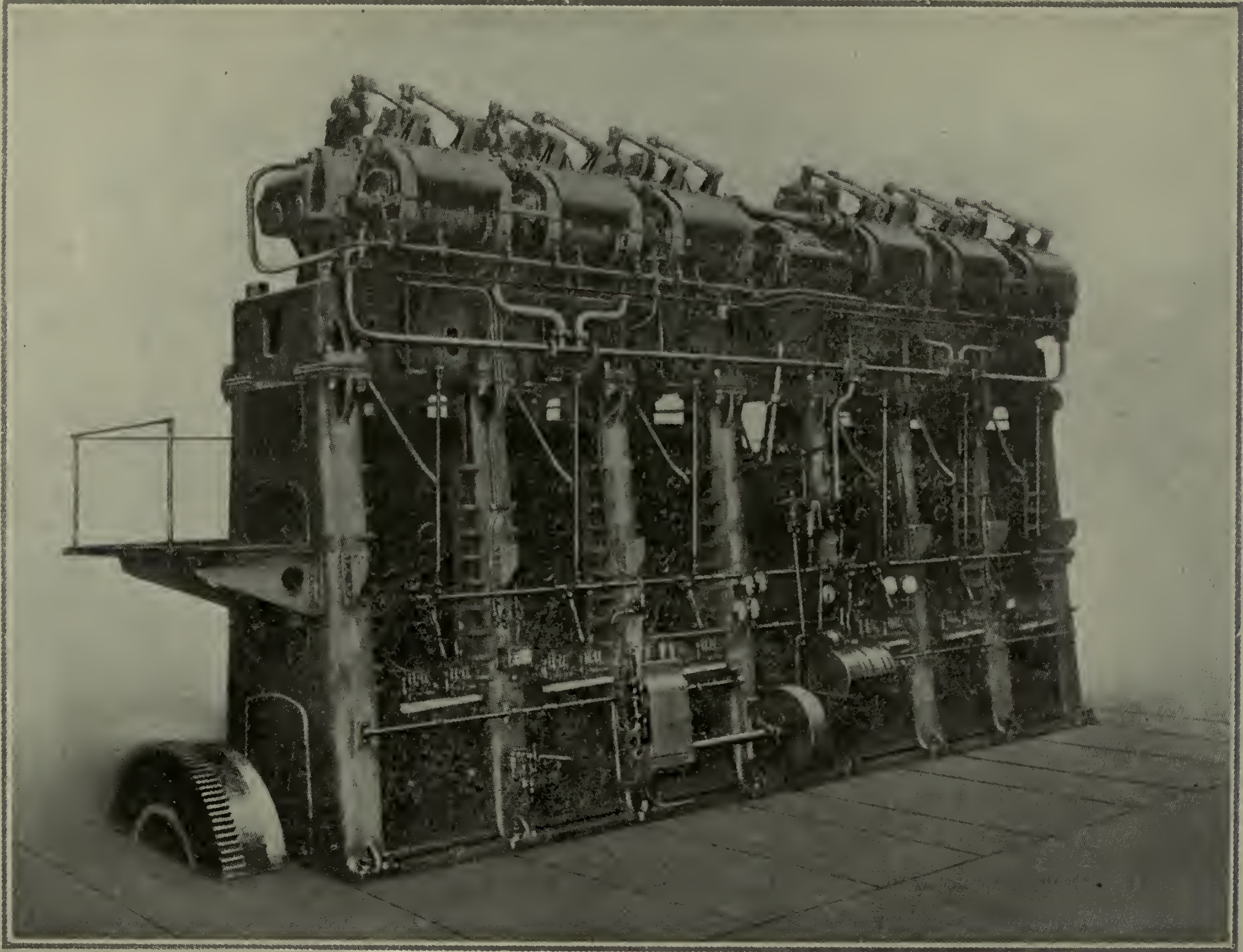


Fig. 78 Single acting four-cycle Diesel Engine developing 1.250 h.p. at 118 r.p.m.

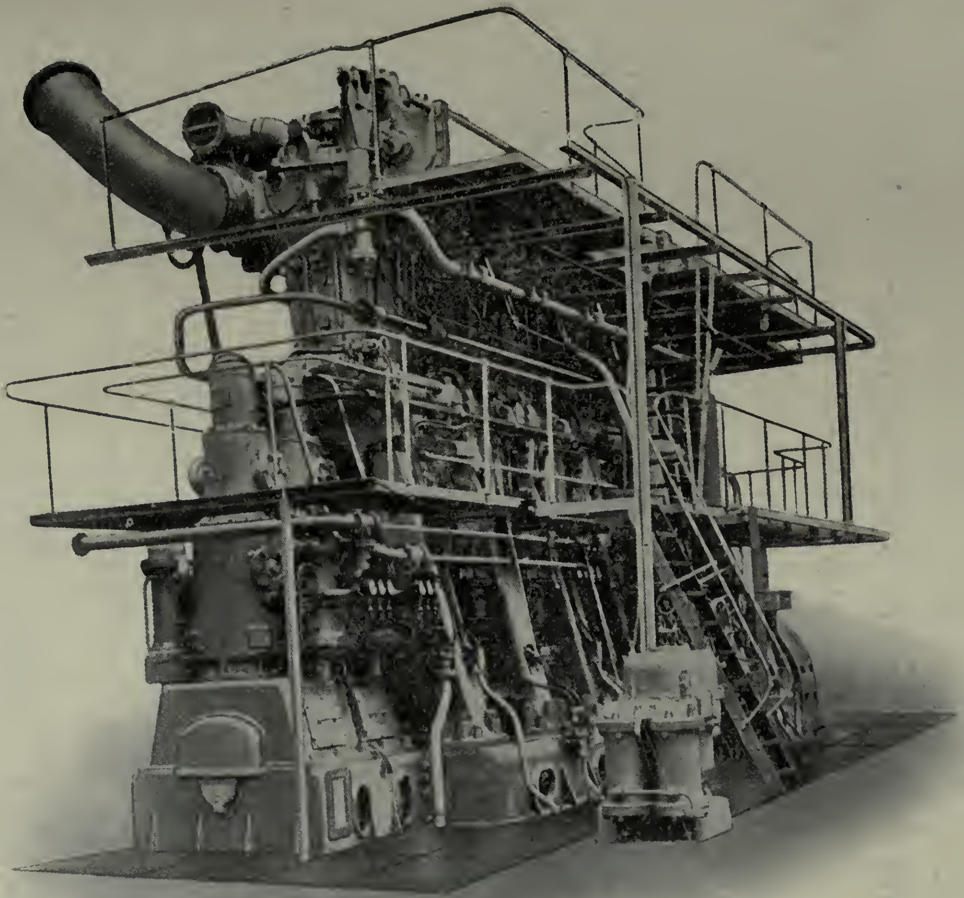


Fig. 79. Four-cycle Marine Diesel Engine developing 1,260 h.p. at 125 r.p.m.

SESSION 1919-20.

PRESIDENT: LORD WEIR OF EASTWOOD.

Notes on the Care and Maintenance of the Solid Injection Diesel Engine.

BY MR. DAVID P. PEEL (Member).

Read Tuesday, December 23, 1919.

The CHAIRMAN: To-night we have a paper by Mr. Peel on the "Care and Maintenance of the Solid Injection Diesel Engine," and as both this paper and Mr. Baxter's on "The Development of the Internal Combustion Engine" deal with oil engines they can be discussed together.

—o—

MR. PEEL.

THE following consists of some notes, based on the writer's experience with the Vickers' Solid Injection Submarine Diesel Engine. The different points will be taken in the following order:—

1, Smoking; 2, Knocking in the Cylinders; 3, Bearing Lubrication; 4, Cylinder Lubrication; 5, The Governor; 6, Water Cooling; 7. Fuel; 8, Setting Sprayer Valves; 9, Diagram of Operations.

Smoking.—There is no doubt that one of the greatest evils to contend with in this class of engine is smoking. the causes of which may be noted as leading to white or black smoke, the latter being more in evidence.

White Smoke is caused by too much air, and is easily stopped by partly closing the air inlets or louvres in the air supply trunks.

Black Smoke is more troublesome, as it may be caused in so many different ways, and consequently much more difficult to locate the cause. It may be due to one or other of the following:—

(1) Too much lubricating oil supplied to the cylinders. When this occurs there is a tendency for the oil to deposit on the piston head, where, owing to the heat of compression it partially vaporises, and the cylinder already having had its full charge of fuel thus the added vapour being unconsumed passes through the exhaust as thick black smoke. The remedy for this is to cut down the lubricating oil carefully to the lowest point of safety.

(2) Where insufficient air is supplied to the cylinders the burning mixture will not be in the correct proportion for complete combustion when the fuel is injected, with the result that a certain quantity of fuel, although vaporised, cannot be burnt and is passed off through the exhaust as black smoke. One cause of this may be the throttling of the air supply by the louves, or again, the passage of air may be restricted thus:—

Where there are sharp bends in the air line, any dirt or dust, which may be drawn in with the induction, tends to lodge in these bends and, in time, forms a bar to the ingress of the air, as in Fig. 1; and thus, the effective area of the inlet pipe is materially reduced.

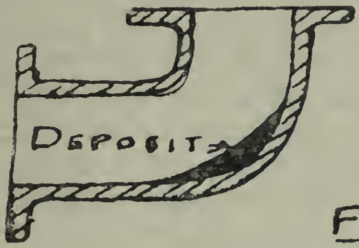


FIG. 1.

This trouble is particularly prevalent where a pipe is led from the crank pits to the induction line, for the purpose of clearing the pits of any accumulation of explosive gas which may be generated by the running heat tending to vaporise the lubricating oil.

The air drawn in this way is generally of a greasy nature, and so assists to coat the pipes with dust which would otherwise pass right through the engine.

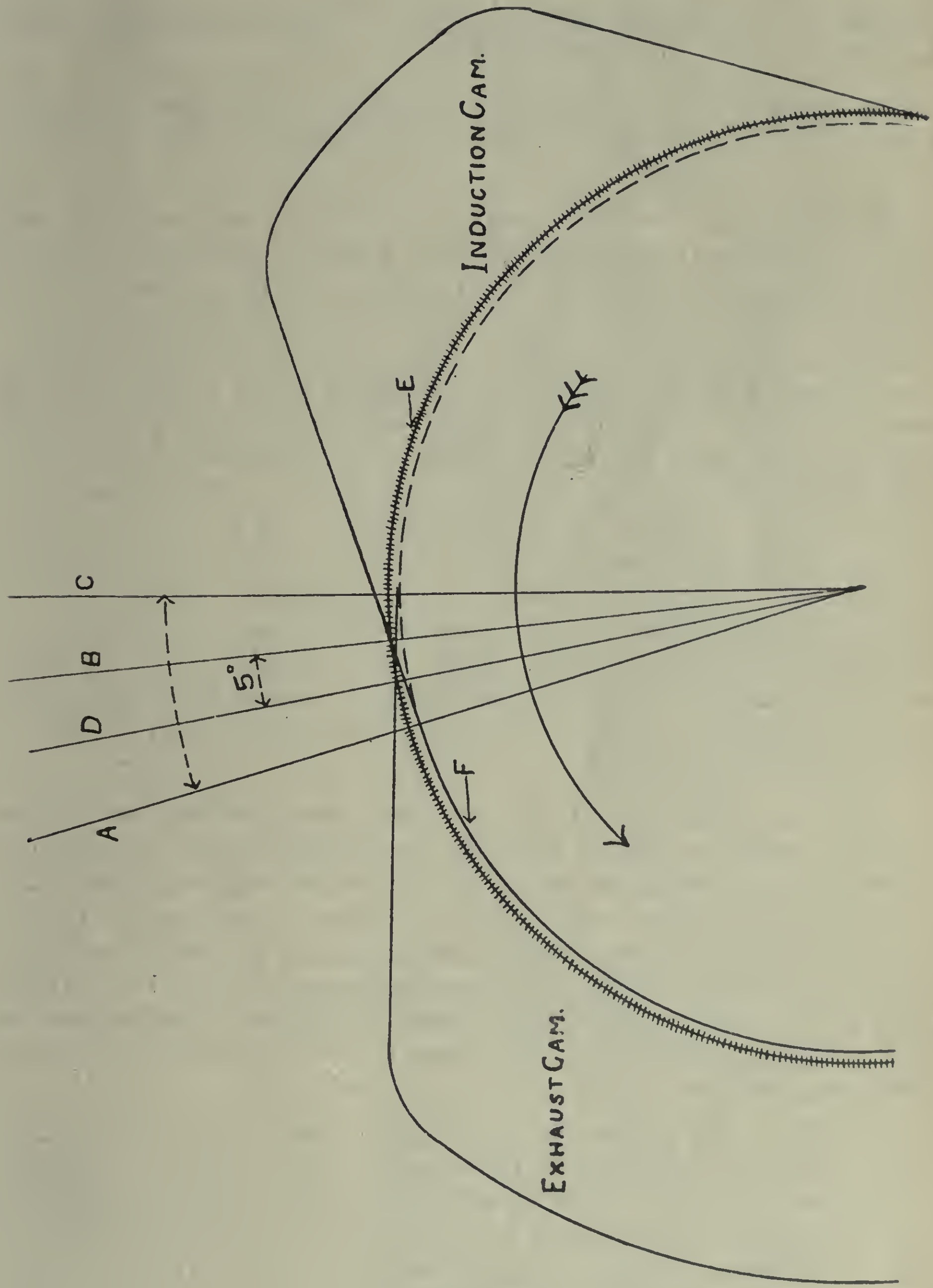
A grid or gauze is usually fitted in this crank pit induction pipe, but it is not sufficient to thoroughly cleanse the air in its passage through the inlet.

(3) If the fuel injection pressure is too low, then the oil, in its passage through the sprayer is not properly pulverised, and so, not in a condition to burn readily, consequently a certain quantity of fuel just passes through the engine in a partially vaporised state, and is discharged as black smoke, so also if the pressure is too high a larger quantity of fuel is injected than can be burnt, with the same result. The pressure to give best results is about 3,000 lbs. to 3,500 lbs. per square inch. The correct pressure is obtained by adjusting the lift of the fuel pump suction valve.

(4) A leaking sprayer valve has practically the same result as too high fuel pressure, and can be rectified by grinding in the fuel valve. The best grinding mixture for this purpose is smooth carborundum paste. When testing a sprayer valve for leakage, a maximum leak of one drop per minute at 5,000 lbs. per square inch is permissible, but it is quite possible to get a valve tighter, still within this limit, it will give satisfactory running.

(5) The following has been cited to me also as a cause:—"If the maker's clearances were altered in the adjustment of the induction and exhaust valve lever push rods, there is a liability to smoking," but no reason was assigned. I investigated this on the engine I had charge of, and came to the conclusion, which may be open to argument:—that the clearance between the push rod roller and cam on this gear averages 30/1000 to 35/1000, supposing these valves are set at 20/1000 instead, so as to reduce clash of gear, then, each cam will pick up the push rod so many degrees before time and drop it so many degrees after time, considering the fact that when the rods are set at correct clearance, the difference between the exhaust valve closing and the induction valve opening, is only five degrees, it will be easily understood when the rods are set at 20/1000 instead of 30/1000 the induction valve will be opening before the exhaust valve is shut, and the engine being on the suction stroke, a certain amount of the exhaust gas is drawn back into the cylinders (Sketch 2), and being unfit for further combustion, occupies space in the cylinder that ought to be filled with pure air, and as the cylinder is still getting its full charge of fuel we have incomplete combustion, the unburnt fuel passing off as smoke. This is illustrated in Sketch 2.

Circle E, is the clearance exaggerated with the radial line D, marking on the clearance circle the point where the exhaust



Sketch 2.

valve should shut, and the line B, marking where the induction valve should open. If we cut down the clearance until the roller is on the line F, instead of the induction valve starting to open at B, it is starting to open at A, while the exhaust valve has still to go from A to C to shut.

Knocking in the Cylinders.—At times we may hear alarming knocks in the engine, due to one or other of the following causes:—

(1) The packing of the sprayer valve may become hard and tight, gripping the spindle so that it cannot return to its seat after being raised by the cam, thus supplying fuel to the cylinder all the cycle. This will not do any damage on the exhaust and induction strokes, but on the compression stroke, the charge is fired immediately the flash point is reached, the result being an enormous premature explosion as this point is reached some time before the proper fuel injection time. This explosion puts a severe strain on the engine, not only towards reversing the direction of rotation against the normal, but also endangering the cylinder cover. The defective cylinder must be immediately cut off, before serious damage is done, then slack back the gland of the sprayer valve, and get free movement on the spindle, after which the cylinder may be reinstated. Sometimes it is not necessary to overhaul the gland, as a few minutes idle running without the oil pressure in the gland will correct the trouble.

(2) The weakening of the sprayer valve spring will give similar trouble, as when the valve is lifted by the cam, the oil pressure in the sprayer acting on the under side of the valve, will not allow the valve to return to its seat, the pressure of the oil being greater than that of the spring in its weakened condition. The remedy is to increase the compression of the spring, or fit a new one. In the former case, care must be taken not to choke the spring as the valve gear might be damaged by too much compression.

(3) Where the engine is so designed that when the exhaust and induction valves are full open, these project into the cylinder within the stroke of the piston and there is sometimes trouble owing to a deposit of carbon forming on the exhaust valve spindle, thus the spindle becomes so stiff in the bush that the spring is unable to return the valve to its seat after being opened, and the valve head is struck by the piston on reaching the end of the stroke, and knocked violently to its seat. A little paraffin placed around the valve spindle will free it from the deposit and overcome this trouble. Fuel will suffice if paraffin

is unobtainable; on no account must lubricating oil be used, as this aggravates the trouble. A weak spring is another cause of this knocking, making the action of the valve too slow.

(4) Too liberal cylinder lubrication is another source of mischief, as a certain quantity of the oil strays past the piston, and is vaporised and fired with the charge, causing a severe explosion.

Lubrication (*Bearing*).—Any engineer will readily understand the vital importance of efficient lubrication, as a high-speed Diesel engine will not run many minutes if the lubricating system fails. The engineer ought to know the system so well that he can locate a fault at once, and perhaps save stoppage of the engine. By knowledge and study, an engineer is often able to detect a coming fault and anticipate it. In the bearing oil system there are usually three sets of strainers, and two gauges, one strainer in the end of the crank pit strains the oil on its return to the oil drain tank. A second is in the oil pump suction line between the tank and the pump suction valves. A third is on the delivery side of the pump to the engine. One gauge is combined pressure and vacuum, and is tapped off the line between the suction strainers and the pump. The other gauge, pressure only, is tapped off the line between the discharge strainers and the engine. By observation of these gauges and a knowledge of where they are tapped from, one can tell what is happening at any point of the oil system. As for instance, knowing there is plenty of oil in the drain tank, on seeing both gauges register below normal, one may conclude the drain tank strainer is fouling, and immediately take steps to clear it. Should the vacuum gauge show higher than normal while the discharge pressure remains correct. then the suction strainers are becoming clogged, and it so follows, if the vacuum gauge is normal and the pressure is low, then the discharge strainers require attention. If both gauges show low and all strainers have been cleaned, then sound the oil drain tank, as a quantity of oil may have been lost through leakage. Some engines are fitted with a valve on the drain tank strainer, to trap the oil in the crank pit. that the drain tank may be cleaned. This valve is apt to be overlooked when starting up, and after a few minutes running the oil pressure falls to zero as the drain tank is emptied into the crank pit; this may lead to shutting down the engine if the engineer is not immediately able to trace the cause of the loss.

NOTES ON THE CARE & MAINTENANCE OF THE SOLID INJECTION DIESEL ENGINE.

Cylinder Lubrication.—There are many methods of cylinder lubrication, two of which I will deal with.

(1) Where a separate small pump is fitted for each cylinder and actuated by a small cam on a roller let into the ram head. The cam forces the ram down on the pressure stroke, a spring returning it on the suction stroke. According to the number of cylinders, so are the pumps arranged to work by suitable means from the main cam shaft. With these pumps the main trouble is the breaking of the return springs, in which event the ram remains at the bottom of the stroke. The pump is so fitted that a spring may be renewed whilst the engine is running. This class of pump requires a good deal of attention as the valves are so small that the slightest particle of grit on the seat will put the pump out of action.

(2) An ordinary rotary cog pump may be fitted to supply oil to the cylinders, through sight-feed lubricators fitted on each one and practically the only trouble is the side wear of the cogs with consequent diminution of pressure.

The Governor.—In some classes of governored engines, where all the cylinder heads are coupled solid with each other and the governor is fitted on the end of the engine, supported by the end cylinder—I have found that owing to the unequal expansion of the cylinder tops relatively to the bedplate when the running heat is attained, the governor shaft, if driven off the main shaft by bevel gearing, is liable to be thrown out of line, causing jambing of the bevel wheels, and gripping of the governor shaft in its bearings, resulting in damage to the bevel wheels and excessive overheating of the bearings. This may be overcome by scraping the bearings on the angle sufficient to allow for free movement when the engine is hot. This trouble is not experienced where the governor is in the centre of the engine and where the cylinders are quite separate.

The Water Cooling arrangement gives little or no trouble, but as a precaution in case of a leak into a cylinder, it is advisable before starting to turn the engine at least two complete revolutions by hand, or dead slow on compressed air, with all explosion cocks open, to see that all cylinders are clear. If water issues from an explosion cock, turning by hand will effect the desired end. No attempt should be made to start while there is the slightest suspicion of water or when the water system is charged with air, as there is a possibility of steam locks forming and stopping the flow of cooling water which must be guarded against.

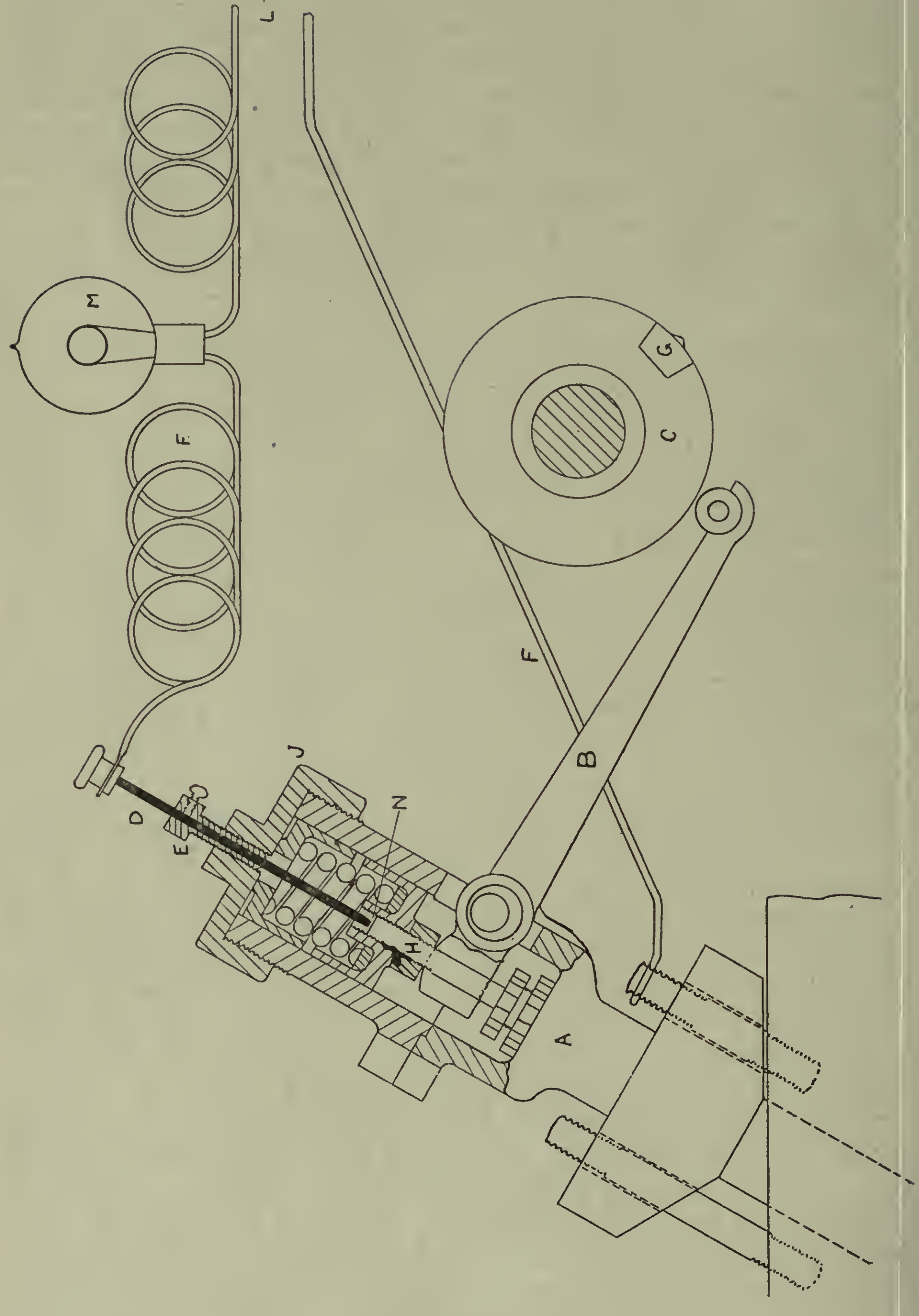
NOTES ON THE CARE & MAINTENANCE OF THE SOLID INJECTION DIESEL ENGINE.

Fuel.—In the case of marine Diesel engines there is a possibility of getting water into the fuel tanks owing to the straining of the ship's plating in bad weather. It is best to keep a sharp eye on the gauge glass of the fuel gravity tank, and if any water shows, open the drain cock at once; this will assist to keep the water below the level of the engine supply pipe, which is fitted well up the tank, just long enough to change over to another tank, after which the first must be cleared.

Timing Sprayer Valves.—There are two methods of setting the sprayer valves:—

(1) Disconnect the fuel supply pipe to the sprayer valve, and then connect a flexible hose to the air starting bottles. Take out the explosion cock and turn the air pressure to the sprayer valve. When trying No. 1 sprayer engine fully retarded, that is, starting to open 4° before the top centre, have the engine turned by hand in the direction of rotation dead slow, listen at the explosion cock hole, and the instant the sprayer valve is heard to blow cease turning, and note on the flywheel graduations the number of degrees between the mark on the flywheel top centre No. 1 and the pointer fixed on the frame marked top dead centre, this number of degrees should tally with the 4°, if not, the toe piece on the sprayer valve cam must be moved forward or backward as the case requires, and the process gone through again until the correct point is attained.

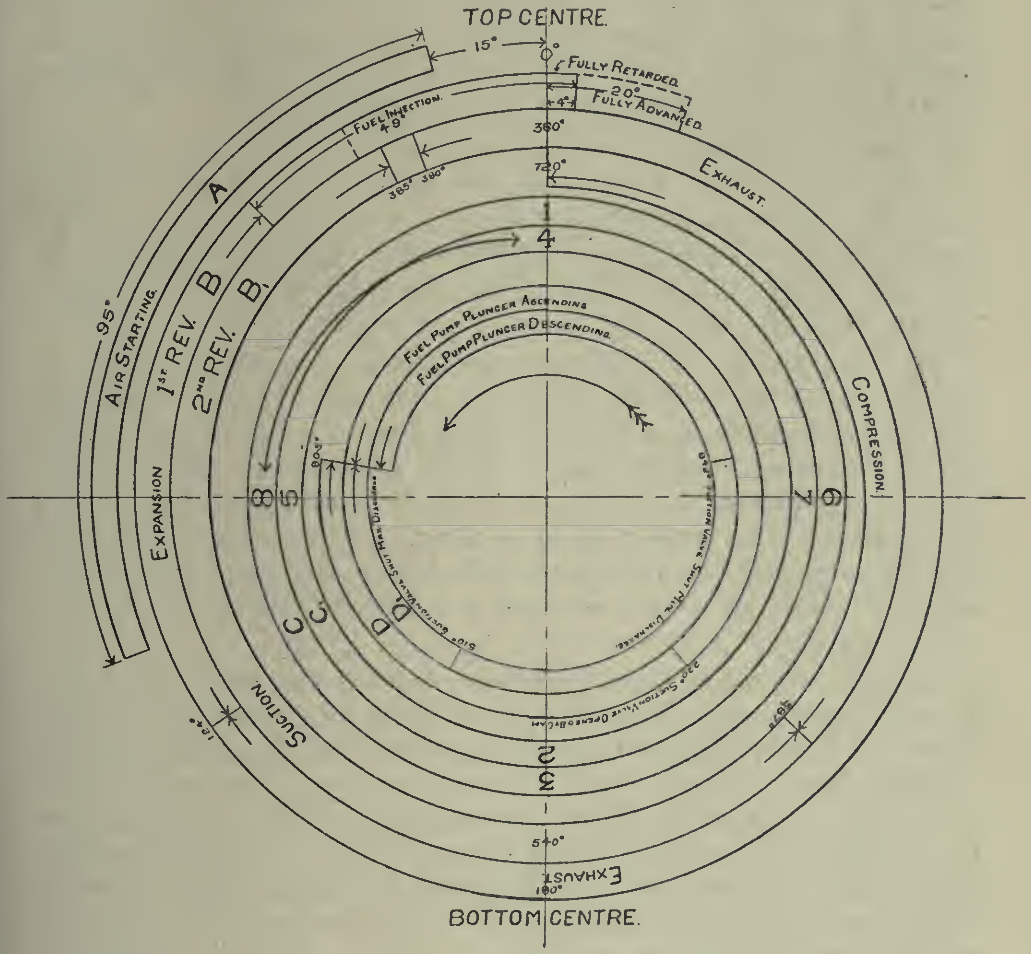
(2) Sketch 3 illustrates where A is the sprayer valve complete, B is the actuating lever, and C the cam with toe piece G; H is the valve spindle. The brass rod D is led through the cap J and insulated from it with the sleeve E made of non-conducting material, and screwed through the cap for fine adjustment. The rod D is set as close to the spindle H as possible without allowing an electric current to jump the gap. An electric wire F is led from the rod D and from the body of the valve to a lighting circuit or battery, with the lamp M in the circuit. The circuit is now complete except for the gap N. Let the engine now be turned by hand dead slow, and as soon as the toe piece G picks up the roller on the lever B the gap N will be closed and the circuit completed, lighting the lamp M, then the flywheel graduations are noted as previously with the air. The disadvantage of the first system is the loss of starting air; the second is not so accurate, as the valve has to be raised a certain amount to close the gap before the lamp will light, when one degree or perhaps two will not be recorded.



Sketch 3.

NOTES ON THE CARE & MAINTENANCE OF THE SOLID INJECTION DIESEL ENGINE.

Sketch 4 comprises a diagram of: A, Air starting. B B Running cycle. C C$_1$, Sequence of firing D D$_1$, Cycle of fuel pump of Vickers 100 h.p. unit engine with eight cylinders.



Sketch 4.

DISCUSSION.

The Chairman: Many ships are now burning oil instead of coal to generate steam, but when one considers that the consumption of oil to do this is about three times the amount required for an equivalent horse-power developed in an internal combustion engine, it is reasonable to conclude that the number of such engines used in the future will increase.

Owing to saving in weight of machinery but principally owing to the lesser weight of fuel to be carried for an equal distance travelled, a ship can carry considerably more cargo when fitted with internal combustion engines, and this is another reason why this type of engine is bound to be developed in the future.

At the present time there is a limit of size for oil engines, but this I believe will be increased. We are apt to compare oil engines which are in a state of development with steam engines which are the result of many years' experience. There were plenty of troubles with steam machinery and we still have a few, and we must expect to have others whilst progressing with the oil engine. The solving of difficulties and avoidance of the cause of these in future designs are factors which assist us to improve.

I shall be glad to know whether there is any limit of size of the cylinder to which can be fitted solid injection of fuel, because we would like to dispense with air compressors for Diesel engines, but I suppose we will have to retain the compressors for making sufficient air under pressure to start or manœuvre the engines. We have found that the majority of difficulties occur whilst the engines are being manœuvred in or out of ports, and there is a possible solution of this difficulty by the adoption of the "Electrification of Ships." If the Diesel engine was coupled electrically to the propeller's shaft we could leave the engine running after stopping the propeller. Electrification also appears to be one way of increasing the horse-power by Diesel engines. For example we might have four Diesel engines each developing 5,000 h.p. with two coupled to each propeller shaft of a twin-screw ship.

In regard to cylinder lubrication it is now common practice to cool the pistons of marine sets with water rather than with oil. We commenced with oil-cooled pistons, and with the bottoms of the cylinders open to the crank case, but we found the consumption of lubricating oil heavy through carbonisation; later we had distance pieces fitted between the crank case and the cylinders and now the pistons are water-cooled and the under sides of these are open to the engine room so that it is known immediately when a piston is leaking. Each cylinder has its own sight feed lubricator.

Marine Engineers will be indebted to oil engine designers if they will give us more accessibility. We do not like having to take part after part away before we get to the one we desire to adjust or overhaul. Simplicity for overhaul saves time and money.

A Correspondent: In his "Development of the Internal Combustion Engine" Mr. Baxter, on page 449, when dealing with the starting problem, refers to the automobile practice of turning the flywheel by means of an electric motor supplied with current from an accumulator. It is true that this system generally works fairly well on land, but it has two inherent drawbacks, the one, mechanical, the other, electric. The mechanical drawback is one of construction, the motor being necessarily small for purposes of economy in weight and cost, has to revolve at high speed to give out sufficient torque to crank the engine, consequently the ratio of pinion to gear wheel must be high. thus necessitating a fine pitch of teeth, if more than one of them is to be in gear, thus we get a flywheel with a number of very fine teeth on its periphery, in a position subject to damage. There is also the certainty that the spaces between the roots of the teeth in the wheel may become choked with dirt and dry grease, absorbing so much power that the motor fails to attain the speed necessary for firing the charge, under these conditions short circuiting will probably take place and the battery be rapidly discharged. it being then necessary to resort to hand cranking, meanwhile the battery is partially run down. and is not efficient for lights or horn or for sparking (if the battery system be fitted for this). The above are some of the serious objections to electric starting gear in the closed-in bonnet of a motor car. On board ship and more especially in half-open and open-boats they are accentuated, and in the open-boats there is the further risk of failure from water damaging the circuits and terminals.

From the above it seems clear that the more compressed air can be utilised for starting and reversing the better for efficiency. The absolute certainty that the desired movements of the propeller can be obained as, and when, required, is of paramount importance on board ship. Reservoirs containing compressed air have been in use in oil engines for probably more than 20 years. I used them in some 25 B.H.P. Hornsby-Ackroyd oil engines installed as power and lighting plant, at Polesden Lacey, then the property of the late Sir Clinton Dawkins in 1903. The installation in duplicate was fully described in *The Engineer* in 1907, Vol. 103, under the heading of "The Engineering of a Country House." These engines, after warming up with a lamp, when stopped in a proper position, were started without any other help, by turning on the compressed air from the reservoir; the reservoir was made in Antwerp, and arrived charged to 150 lbs. Pressure was maintained by a small air pump worked by the engines when required. On a vessel it would be necessary to provide an efficient hand-pump as an additional precaution, such pump if on the triple compression system with three cylinders as made by Messrs. Hatterley and Davidson, Sheffield, for tyre purposes, is easy to work.

In regard to silencing I have found that the gradual increase of the diameter of the successive lengths of the exhaust pipe effects this. On board ship the weight precludes any great length of pipe, but I believe something might be done on the principle of the musical instrument known as ophicleide. A pipe increasing continuously in diameter is curved into one or more circles. On board ship these coils must lie in a horizontal plane to avoid water trap. The same coil system, if adopted on the petrol pipe leading from the distant tank to the engine, would probably prevent the fracture of such pipe from vibration, it also provides for expansion and contraction. Mr. Sopwith, to whom I recently suggested it, told me that he has used it for petrol pipes for several years with complete success in his aeroplanes.

In regard to fuel economy, much can be done by the utilisation of some of the heat in the exhaust gases. Four years since I fitted to the engines of my Overland car an air-heater of my own design. A concentric sheet steel casing fits over the horizontal length of the exhaust pipe from end to end, this casing is hinged at the top for easy fitting, nearly airtight over each of the four exhaust duct branches, a further casing is fitted to the exhaust pipe after its downward bend, the concentric casings thus formed communicate with the induction pipe a few inches above the carburetter, so as not to heat it. A plug cock controlled from the steering pillar regulates the supply of hot air which attains a temperature of 400° F., raising the temperature of the whole mixed vapour by over 100 degrees.

This apparatus was covered by a patent I took out in 1894, it increased my mileage per gallon from 15 to over 18, even on our bad roads. I can go up most hills on top speed, and sparking-plug troubles have ceased, for it is easy to secure a reducing flame, by which all carbon deposits are burnt off, if they form, it appears that this arrangement, wherever possible, would be equally efficient in marine motors. I have found in heavy sea-fogs near the coast that the turning on of the hot-air cock, immediately gave results as good as on the driest day in summer. It is now always in use, and has caused no trouble with lubrication from the highest temperature of combustion (if the heavier oil such as the Vacuum Co.'s B be used) the apparatus does for the internal engine what feed heating and superheating do for the steam engine. The cost in my case was £2 10s. out of pocket, and my own time fitting it on and making the link and lever connections.

A Member: I have not an intimate acquaintance with the solid injection engine as described by Mr. Peel, but can claim to possess considerable knowledge of the Diesel engine as applied to German submarines. I think the solid injection engine is unlikely to be a commercial success except possibly in small sizes. Undoubtedly its system of securing pulverisation of the fuel is highly efficient but there is an absence of turbulence. The oil is injected into the cylinder and remains more or less stationary in the combustion space. This militates against the attainment of a high M.E.P. with reasonably moderate maximum cylinder pressure and good combustion. On the other hand in the Diesel engine, the fuel injection into the cylinder is followed by a blast of air which swirls the fuel in the combustion space, ensuring close inter-mixing of fuel and air, admitting of a bigger output per unit of cylinder capacity. That is the difference between the solid injection engine as used in British submarines during the war, and the pure Diesel as adopted by Germany.

Apparently at the commencement of the war, there were five or six different makers supplying Diesel engines to the German Admiralty. Towards the end of the war it would appear that Germany relied entirely on two makers only, namely, Krupp for two-stroke engines and M.A.N. for four-stroke units. The two-stroke machines appear to have been favoured mainly for short radius of action, *e.g.*, work in the North Sea and Mediterranean, and reliance was placed on the four-stroke boats for a more extended radius of action. The former are lighter, power for power, and the earlier units were constructed largely of manganese bronze and aluminium in place of cast iron, though as the war progressed, there is ample evidence supplied by Germany's shortage of these materials because cast steel replaces manganese bronze crank cases and frames in the later two-stroke units.

The four-stroke units throughout the war were mainly constructed of cast steel, and as the war progressed one surveys with interest the efforts made by the Germans to secure a greater horse-power per unit of cylinder volume. Thus, in the latest designs two fuel valves per cylinder are employed to augment the M.E.P. and pistons so small as 15in. bore are oil cooled. Likewise, in the larger units, main bearings are cooled to admit of running the sets at the very high piston speed of 1,300 feet a minute. There is evidence to show that engines were not often run at this terrific speed, and probably the Germans only resorted to it in time of dire emergency.

There is a marked difference between English and German practice of building the engine into the hull. In the English submarines the hull is made virtually part and parcel of the engine and the latter cannot easily be dissociated from the hull. In the German boats the engine is quite a self-contained unit with a substantial bedplate bolted down to suitable joists in the hull. It has therefore been possible to remove German engines from the boat and apply them industrially for land work, and I very much doubt if it will be commercially feasible to apply English engines similarly.

During the year 1919 my firm has taken out 40 or 50 German engines and are engaged on the erection of a large number for generating electricity. The experience so gained convinces me that in the four years of the war Germany advanced Diesel engine design more than would have been accomplished in 20 years of peace. One of these sets in particular, appears to have been used experimentally, for there were a number of devices fitted, the object of which could not immediately be seen. For example, a little thing which puzzled us for a long time was a branch from the exhaust silencer to the air suction to the cylinders, by which a proportion of exhaust gases could be admitted to the cylinders or entirely cut off at will. I think the explanation of this device is to reduce the noise of the exhaust when travelling on the surface at cruising speed, for the reason that when we were experimenting on land engines to burn tar oil, we found that the admission of hot exhaust gases to the cylinder when the engine was running at small loads, not only allowed the engine to run more steadily, but also greatly reduced the noise of the exhaust.

The Chairman: In regard to solid injection. Is it a question of weight for the submarine engine?

Member: The object of the designer is to secure the maximum power per unit of cylinder volume and with the minimum weight.

Those who advocate the solid injection engine generally refer to the extra complication of the Diesel Engine bv virtue of its having a compressed air plant attached thereto. I think there is very little foundation for this complaint. A properly designed Reavell compressor can operate 4,000 hours without any examination whatever. Certain other compressors require much more frequent examination and their valves have to be cleaned every 200 hours or there about. Hence, by employing a really good compressor, and I am very partial to one of the Reavell types, the extra complication gives no cause for complaint.

The Chairman: There is a certain amount of weight added by installing the compressor?

Member: On the other hand the better combustion secured by the air pulverisation and turbulence consequent upon the inclusion of the compressor admits of the adoption of a higher power output which reduces the specific weight of the engine.

The solid injection engine appears to have reached its present useful limit in the 14in. cylinder. Even so the indicator card is very peaky and its reliable M.E.P. in the 14in. cylinder is probably round about 90 lbs. per square inch. In the case of the 17in. cylinders on the monitors, there was a great difficulty experienced and such an engine is possibly fullv rated at 65 lbs. per square inch M.E.P., and even then the indicator card is excessively peaky. One of the artificers on board told me they were greatly troubled with seized pistons and often had to resort to the hose pipe to cool them down. On the other hand, the Germans appear to have evolved a design of Diesel engine which has worked, at least for short periods, at a mean effective pressure ranging round 120 lbs. per square inch.

In any case the solid injection engine is quite untried for commercial purposes and is undeveloped, nor do I think it has a commercial future, except in quite small units.

A Member: I take considerable interest in Internal Combustion Engines, and with your permission, would like to say a few words with reference to both Mr. Baxter's and Mr. Peel's papers.

Looking at the general development of oil engines you will find that there is a tendency for the so-called Semi-Diesel engine and the Diesel engine to meet each other inasmuch as the former is employing gradually higher pressures, whereas with the latter it has in recent times been attempted to reduce same. In the initial stage of the development of the Semi-Diesel engine, the hot bulb represented a factor of ignorance on the part of the designer inasmuch as he was incapable of designing a cylinder head suitable to withstand varying temperatures. Water injection was added to prevent the hot bulb from attaining too high a temperature at full load and the advantages of water injection were advocated most extensively by all manufacturers. To-day with the advanced knowledge of the designer the hot bulb as an uncooled portion of the cylinder has disappeared and with it the water injection system which is now admitted to be an inconvenient and obsolete design.

If you want to atomise, pulverise or vapourise any fuel satisfactorily, one of the essential factors is heat, and by removing the uncooled hot bulb the designer was faced with the alternative of having to increase the compression pressure.

With the Diesel engine in which already high pressures have been utilised it has so far been found necessary to inject the fuel by means of compressed air in order to get satisfactory atomisation. What the hot bulb is for the Semi-Diesel, the air compressor is for the Diesel engine and these weaknesses have been recognised. There is now a tendency to design an engine which will dispense with the air injection system of the fuel and utilise the same method as in the Semi-Diesel engine, *i.e.*, the solid injection principle. It is stated that solid injection will never be capable of atomising the fuel as finely as air injection but my actual practical experience has already convinced me of the contrary. I have had the privilege of attending tests of some of the latest solid injection engines which, I am pleased to say are purely of British origin, and find that they are not only capable of atomising any fuel satisfactorily, but that they are even capable of atomising heavier fuels than the present design of Diesel engines employing air injection. With air injection each particle of oil must be surrounded by particles of air and consequently on entering the cylinder the heat of compression must be imparted to the globules of oil through the surrounding layer of air. The more finely the oil is atomised the more important becomes the time limit for passing on the required amount of heat to each globule. The previous speaker has expressed himself against solid injection. He has invented a most excellent pulveriser, known as the Sleeve Pulveriser, in connection with which he attaches great importance to the sharp edged flame plate, and I consider that his system is a combination of mechanical atomisation and air injection atomisation. Again take the present method of burning Tar Oils by means of pilot ignition. A pilot jet is arranged in such a manner in the fuel valve casing that there is no possibility of it coming in contact with the air and consequently atomisation of the pilot jet is purely mechanical. Of course, it should be admitted that, the pilot ignition oil being of very good quality, very little atomisation is required for its complete burning, but it shows that the designer is quite aware of the retarding effect of the atomising air on the transference of the heat from the highly compressed air to the globules of oil.

The question of accessories is of greater importance on board a ship than in land installations. A plant which requires very close supervision—may be, it is technically a very sound proposition—is not considered a running job by the marine engineer. There is no question that the least number of accessories are an essential condition and every one of the various accessories which a designer can dispense with will help to popularise the internal combustion engine with marine engineers.

The opponent to the solid injection system will say, this is all very well, but what about your high fuel pump pressure. In my opinion it is easier to design a fuel pump for a working pressure of 3,000 per square inch than a three or four stage air pump working at 1,000 lbs. per square inch.

Mr. Peel does not say anything about the question of fuel. It has been stated against the Vickers' engine that unless adjusted very carefully it will give a smoky exhaust. It is admitted that conditions in that direction are not perfect, but it is only a matter of experiment, and there are in fact solid injection engines which can burn any fuel such as used under boilers in a most satisfactory manner. To-day with the high price of labour it becomes a question of material, and it is absolutely essential that an engine should be capable of burning any grade of oil without losing in efficiency. If you have to go into a port and spend say, a day in adjusting your fuel pumps to suit any particular oil such a state of affairs is not satisfactory and the position to-day is that we must adapt the engine to suit available fuels and unless an engine is flexible with regard to burning any kind of fuel it cannot be considered a commercial proposition.

A Member: I would like Mr. Baxter to enlighten us on several points in connection with his paper. The marine engineer is, I think, conversant with the ordinary cycles of the internal combustion engine. Mr. Baxter asks us to note a cycle, a diagram of which is shown in the paper, and in connection with this I was in doubt about its value. I have, however, got an explanation from a friend, who writes as follows:—

The cycle suggested is simply the constant pressure cycle of thermodynamics in which expansion is carried down to atmospheric pressure, but modified by the creation of a vacuum from B to C and back again from C to B.

This vacuum retards the piston at the end of the admission stroke and similarly helps it forward at the beginning of the compression stroke, so that it does not directly bring about any net loss or gain of energy.

The thermodynamic efficiency of the cycle in which expansion is carried down to atmospheric pressure is certainly rather higher than the usual constant pressure cycle in which expansion is only carried as far as the point F, but the actual practical efficiency is, however, decidedly less, for the reason that the mean effective pressure is less than half. owing to the very low average pressure above atmosphere of the part of the diagram F B G.

Dugald Clerk has compared the two constant pressure air cycles—

(*a*) With expansion carried to atmospheric pressure.

(*b*) With expansion carried only to the point where compression starts, *e.g.*, point B on the Durtnall cycle.

The efficiency of (*a*) is 64 per cent.

The efficiency of (*b*) is 56 per cent.

While M.E.P. of (*a*) is 56 lbs. per square inch, and M.E.P. of (*b*) is 117 lbs. per square inch.

In the above cycles a maximum pressure of 500 lbs. per square inch has been assumed.

To obtain a given power, the size of the engine has thus to be more than doubled and the small thermodynamic saving is more than dissipated in friction, first cost, etc.

I would like Mr. Baxter to enlighten us if that deduction is correct.

Mr. CHARLES BAXTER: A Member discusses the solid injection system versus injection by means of compressed air, and in this particular, I must say that the efficiency of the solid injection engine is fairly high and I see no particular reason why it should not be brought up to that of the Diesel engine. The impression that each globule of oil should be surrounded by a certain indefinite number of atoms of air is more or less a fallacy. There is one point to notice and that is, that so far, with compressed air injection, it appears easier to vary the diagram area than with solid injection. With the solid system we get a peaky diagram, and when it comes to cutting down the injection it is found to be a delicate business.

A MEMBER: Not the latest type.

Mr. BAXTER: My experience is that it needs very fine workmanship and better design than is put into it at present. The ease with which the Diesel engine cuts down its diagram is shown by Fig. 65 in my paper.

Another point raised concerns the term "Semi-Diesel" and I would draw attention to the fact that I pointed out in the paper that this was perhaps an inappropriate term to use.

With regard to the remark by the same member as to certain fittings on a German submarine engine, the object of which it was difficult to discover, the fitting admitting exhaust gas, I would say that I have used the same device. I found that by taking in inert gas the explosion is considerably cushioned and in certain engines I found that I could get rid of knocking pretty well without the use of water, which is often very inconvenient.

The MEMBER: We knew what the object was; but what it achieved was not clear, but on experimenting we found that it reduced the noise of the exhaust appreciably.

Mr. BAXTER: This follows as a matter of course, because the inert gas reduces the diagram area and the engine exhausts nearer atmospheric pressure.

MEMBER: We took curves and could not detect any difference, the exhaust was very quiet.

Mr. BAXTER: To refer to the discussion on my own paper I would first like to add a few words to what I said about combining steam and internal combustion engines. Lately a great deal has been heard of this type, as the "Still" engine is of this variety, but it was by no means the first.

Professor Spooner some years ago constructed an engine upon this principle, which had four valves per cylinder, two steam valves, admission and exhaust, and two for the explosive mixture, inlet and exhaust. On the first outward stroke the engine inducted a charge of mixture, compressing, firing and exhausting it on the following three strokes, as is usual practice. Steam was then admitted on the fifth stroke and exhausted on the sixth.

By these means he constructed an engine in which better economy could be maintained, as from a thermodynamic standpoint the two sides were mutually helpful. The only difference between this and the "Still" engine is that in the "Spooner" engine the steam and combustion cycles took place on the same side of the piston.

The "Spooner" engine was of very simple construction and the patent specification, number 26271 of 1906, is well worth perusal.

With regard to another member's communication, I agree with his remarks in many respects. He speaks of starting engines by an electric motor, and whilst this is not entirely satisfactory for automobiles, it works well up to a point. The difficulty he is up against, due to dirt collecting in the teeth is non-existant in properly designed machines as the gearing is enclosed in the crankcase.

Mr. Peel mentions grinding valves, which is a very important subject. A small steam leak does not make a great deal of difference, but when it comes to internal combustion engines it is absolutely necessary that valves should be tight if the engine is to perform well. The usual thing is to smear the valve seating with coarse emery and oil and rotate the valve in it in a semi-rotatory manner, next using a medium grade and finishing off with fine. Personally, I find that the rough can be cut out unless the seating is very bad, and if the job is very particular, as in racing engines, the finishing can be done with crocus powder. The trouble usually experienced is that small pieces of carbon get hammered into the seatings, a point which can be mitigated by fitting valves with a tungsten content as mentioned in my paper.

When valves have been ground a few times they sink into the seating and in order to remove the top ridge, which detracts from the area of opening, I turn up a valve out of tool steel and cut teeth in the seating, using it as a cutter. Mr. Peel also mentions the influence of tappet clearances upon the period of valve openings and he is quite right upon this point, this is why makers always stipulate the clearances in thousandths of an inch.

The trouble mentioned of sticking injection valves is a point which designers are up against, for to get a valve to delicately function under high pressures is not easy, and when high temperatures are also involved the subject becomes more complex, involving the study of metallurgy. He also mentions the subject of lubrication which has always been a worry to me. Internal combustion engines are not lubricated with the same ease and certainty as steam engines, and no warning, or very little, is given before a seizure takes place which may result in serious damage. For this reason the only system is the completely forced installation which is now in extensive use.

He also speaks of gear pumps: I tested pumps of this type for an output of 10,000 horse-power per week, arranging them to pump oil against 45 lbs. per square inch, obtained by closing a cock on the delivery side. These pumps are very reliable if kept primed by being situated below the oil level, and it was found that the clearances, both end and over the teeth, need not be very fine, for when they were increased from 0·005in. to 0·015in. no appreciable difference was noted in the quantity delivered.

A Member asks a question *re* the cycle proposed by Mr. Durtnall. This cycle from a thermodynamical point of view is efficient as has already been pointed out by Captain Riall Sankey in one of the Society of Arts lectures I believe, but whether the amount gained would be more than counterbalanced by friction is a moot point. It also suffers from the same drawback as the six cycle engine mentioned in my paper.

In this engine the strokes are: induction of mixture, compression, explosion, exhaust, induction of air, exhaust of air; the object being, as explained, to get rid of the dead gas in the unswept portion of the cylinder. This gives an engine of higher theoretical efficiency, but as the power developed is proportional to the number of explosions it will be seen that a larger and heavier engine results. The remarks read by the member are quite true, but I can hardly agree with the view that the extra efficiency gained would be negatived by extra friction, whatever the size of engine produced.

A Member: Would it be useless unless you went into very high powers?

Mr. BAXTER: Yes, substantially-the usefulness of this cycle, if any, would be best apparent in large sizes as the frictional loss is of course less, proportionally.

I think there is very little to say further, except perhaps in connection with the limit table given on page 481, Fig. 37, of my paper.

This is the standard table used and is said to be on a hole basis, which means that all holes drilled or bored are only allowed to depart from the size chosen to the extent of the limit given in portion A and B, A being, for very accurate work, and B for ordinary manufacture; a one inch B class hole, for instance, may be larger by 0·00075in. or smaller by 0·0005in., which is three quarters and half a thousandth respectively.

Shafts, etc., are machined to similar appropriate limits, for instance a shaft to be a running fit in the one-inch hole chosen above would be between 1in. minus 0·00125in. and 1in. minus 0·00275in., in other words between 0·99875 and 0·99725in. (an X class fit). It is quite easy to work to these fine limits by "go and not go" gauges, which are kept in stock for all the sizes used. Registers are push fits, etc., and a good designer with plenty of shop experience can cut out the quarter and half thousandths if the work is of fair proportions, for it is useless fixing limits to half thousandths of say a part 8in. diameter, as it is neither practical nor a commercial possibility and only hinders output.

There are many methods of writing limits on drawings, for instance, where gauges are stocked, a three inch B class hole would simply be dimensioned 3in. B, and the man would procure from the tool store a 3in. B gauge, one end of which will enter the hole and the other end will not. In odd sizes for which there is no gauge, say $3\frac{1}{16}$ in., a B class hole could be dimensioned $3\frac{1}{16}\,\begin{smallmatrix}+ & 1\frac{1}{2}\\ - & \frac{3}{4}\end{smallmatrix}$ in., the limit of course meaning $1\frac{1}{2}$ and $\frac{3}{4}$ thousandths, the man working to a micrometer in this case.

After much experience, however, upon designing work of all descriptions I have no hesitation in saying that I by far prefer the limit being written in the $3\frac{1}{16}\,\begin{smallmatrix}+ & .00150\\ - & .00075\end{smallmatrix}$ form, a square being drawn round the limit portion so that the machinist can readily detect the important sizes, the other dimensions being assumed to be plus or minus ten thousandths, as the case may be, and a note to that effect put on the drawing.

I find it a great advantage to even further improve upon this, keeping any limited dimension in the complete decimal form and for that reason the practice in the office for which I am responsible is to write the limit as shown above, drawing a small square round it and putting the size over the top as a decimal. The works have instructions to work to the uppermost limit in the square and in the case of holes this the draughtsman makes the lower limit and in the case of shafts the higher limit, the percentage of scrap being greatly reduced by these means, as if the machinist takes slightly more off than he intended, the work may still be within the limits set.

It is a great advantage to have a competent view room where work is inspected in different stages to prevent waste of time and money, finishing accurately parts which are already scrapped by say, one of the dimensions being below size.

In connection with cylinder sizes I would say that the difficulty experienced in designing large engines is purely one of arranging the metal round the cylinder so that internal strains are not set up in it, which cause distortion and breakages. There is no reason whatever why, with scientific design, engines should not be built of any size, more than equal to the largest size steam engines in fact, and with suitable arrangements, solid injection.

In conclusion, I would add that the question of cooling was dealt with in my paper, it being pointed out that the cooling of pistons and valves, etc., was now current practice, the question of get-at-ableness was also touched upon, the reason being given for the prevailing inaccessability.

Mr. David P. Peel: With regard to solid and blast injection there is much to be said in favour of both.

With solid injection, owing to the absence of the injection of air at a very low temperature with the fuel the difficulty of starting up in cold weather is greatly lessened. I have known a case where owing to this the entire supply of starting air was consumed in an effort to start, and then to fail in doing so. When starting a multi-cylinder engine many cylinders are on air, turning, and the remainder on fuel. The blast air is about 800 lbs. per sq. in., the compression about 400 lbs. per sq. in. Now in reducing air suddenly from 800 to 400 lbs. sq. in. there is a corresponding reduction of temperature, which in turn has a cooling effect, both on the heat of compression and the fuel, hence the difficulty of starting. Where blast injection has a real advantage is, in the superior possibilities of adjustment with regard to smoking.

The solid injection, with the exception of two stroke, has its air supply limited to the full opening of its induction valve and louvres. The blast injection can receive a further supply from the sprayer valve after the fuel has been injected.

A Member, in the discussion, claims an advantage in the blast system from a point of turbulence in the combustion space. I think this is easily evened up by the superior pulverising in the solid system from this point of view. I have found when a minute leak developed in the pressure side of the system the fuel escaping under a pressure of 3,000 lbs. per sq. in. and upwards came out in the form of vapour and was invariably detected first by sense of smell. This, together with the fact that in the Vicker's 100 h.p. unit, the sprayer nozzle was pierced with only 5—18/1,000 holes, proves the excellence of the pulverising.

Regarding the solid injection engine from a commercial point of view, I would like to mention a little about the class of engine at present being built by Messrs. Doxford, of Sunderland, to whose Chief Engineer I am indebted for the information. This engine is on the Junker principle of opposed pistons in the same cylinder driving three cranks and using solid injection. The experimental engine on the test bed had a range of speed from a minimum of 30 r.p.m. to a maximum of 130 r.p.m., and in one cylinder developed 500 h.p. normal.

Considering the size of the engine this range is distinctly good, and the minimum creates a record I think. By throttling the exhaust, 750 h.p. can be obtained for an hour or more when urgently required.

The same Member mentions the piston speed of German submarines. In the British boats the average is about 990 ft. per min. at ordinary work, but this could be increased to about 1,140ft. per min. We maintained the latter once for fourteen hours until arrival in harbour, bringing home a seriously injured member of the crew.

I have read with interest the point raised about water and oil-cooled pistons. I have not had a great deal to do with these classes, but as far as I have had I can quite corroborate the statement with regard to the loss due to carbonisation.

There is another point against the oil system, that is the retention of a certain amount of heat by the oil due to imperfect cooling, and consequent diminution of efficiency as a cooling agent. However, there is the advantage that in the event of a leak developing the oil merely falls to the crankpit and is not lost. On the other hand a similar leak from water cooling, unless precaution is taken, falls into the pit and is passed round with the bearing oil, and has a bad effect on the efficiency of the pump, if a geared pump, also causing marking of the crank and gudgeon pins if the brasses are inclined to be easy.

I would like to corroborate the statement concerning the method of building the engine in, in the British submarines. The bed-plate consists of two channel girders laid on edge with the main bearings bolted between, the crankpit is formed in the structure of the hull.

A Member asks about fuel not being mentioned in my paper. I will give a little information on this subject with regard to submarines. Up to about November, 1915, Broxburne was used with, generally speaking, very good results, about the only trouble being the carbonising of the piston rings, the top two usually, but sometimes as far as the fourth were found to be solid with the piston.

About November, 1915, American distillate was introduced, my boat (*D*3) being the first to give it a trial. This fuel proved highly successful, an equivalent h.p. was obtained on a lower consumption. After running for a considerable time, a piston was withdrawn for examination, it was found to be coated with a thick heavy grease and the rings perfectly free. The only disadvantage was from the fumes of the cold oil, which in the confined space, and diving for 20 hours per day, were particularly unpleasant, and resulted in sickness amongst the crew. In the steam boats the generator engine was fitted to use boiler fuel in the event of a shortage of distillate, but I have never known of an occasion arising necessitating the use of it.

When speaking of smoking I do not wish to infer this was a prolific source of trouble, but only in the sense of the numerous ways by which it could be caused. In the case of the propelling engines we were fairly clear. It was in the generator engines, as fitted in the steam boats, where we had most trouble. This was due to not being able to alter the fuel pressure, while running, as simply as in the propelling engine, where the simple turning of one handwheel regulated the pressure on the entire engine; in the other, each cylinder rquired to be handled separately, apart from the regulation done by the governor.

My remarks on grinding in valves, as spoken of by Mr. Baxter, were in reference to the sprayer valves only. Coarse emery must not be used on these, the fuel pump valves can also be ground in with smooth carborundum paste, although it is sometimes necessary to use flour emery first. When a fuel pump valve has been leaking it has the appearance of having had a sharp knife drawn radially across the seat.

Mr. Baxter mentions about the efficiency of the geared pump. My experience is briefly this: The geared pump when drowned and pumping new cold lubricating oil for cylinder lubrication through sight-feed lubricators, maintained 150lbs. per sq. in., and when pumping dirty bearing oil and lifting about 12 inches would give a pressure of 15lbs. per sq. inch with the oil cold. As soon as the engine and oil attained running heat we were pleased to get 1½ lbs. per sq. inch.

I have to thank Mr. Baxter for his corroboration of my idea with regard to the effect of tappet clearances on smoking, and also the members for their interest in my paper.

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SESSION 1921-22.

President: SIR JOSEPH P. MACLAY, BART.

Practical Notes on the Installation and Running of Petrol, Petrol-Paraffin and Semi-Diesel Engines.

BY LIEUT-COL. D. P. LAMB (Member).

Read Tuesday, March 1, 1921,

DURING the past few years I have come a good deal in contact with motor craft of various classes, from the fairly high speed motor boat (petrol), which can be used either for business or pleasure, the runabout boat (petrol starting, running on paraffin), which may be seen daily doing service about harbours, docks and shipyards, and the larger type of boat of the coaster class, for carrying cargo, which in some instances are fitted with petrol-paraffin installations, but are more commonly found fitted with a slower running motor of the semi-Diesel type. A considerable number of motor tugs are now fitted with this type of motor.

As each motor is of a distinct type, and is installed into a boat which is considered to be the most suitable for the work it is intended to do, it will be necessary to divide them into sections before proceeding further, and then deal with them in the order named.

1, Petrol; 2, Petrol-Paraffin; 3, Semi-Diesel, Air Injection and Solid Injection.

As each of the above-named types has its own peculiarities due to a variety of causes, many of which only become apparent after the motor has been installed, I will deal with the installation in the first instance.

Installation.—The same general rules apply whether the motor to be installed is of high or low power or whatever type of motor may have been selected. It is generally understood that all motors before being despatched from the makers' works have passed the necessary test bed trials, where each motor is run for a stated time under load, when the B.H.P. developed and fuel and lubricating oil consumption is noted. When it arrives at its destination, where it is to be installed into the boat, it invariably passes into other hands, whether provided by the maker or not is not material so long as they exercise the necessary care when fitting out the installation, for success or otherwise is in a great measure due to such. A good motor may prove very unsatisfactory through being improperly installed. A blue print drawing is usually supplied to the man in charge of the installation showing line of shafting, position of motor and reverse gear, also position of fuel and lubricating oil tanks, exhaust piping, seacock, circulating inlet and discharge pipes, the latter in many cases being led into the exhaust pipe, as near the exhaust manifold on the motor as convenient, when the exhaust pipe is led out through the stern or through the boat's side. Assuming the motor is being installed in a new hull where boat-builders have all he necessary particulars *re* motor seating, etc., one does not expect to find the same troubles crop up as where the hull has been altered to suit the motor.

Line of Shafting.—As the length of shafting is not very long in the various types of motor craft mentioned, a line is usually stretched, passing through centres, which have previously been fixed in the stern post and forward end of engine room, care being taken to see that the line at forward end of engine room coincides with the centre line of the boat, and the height of the line above keelson is correct to the dimension given on the drawing. In all probability the motor seating will be already in place, if so, the position should be checked as to width and height. It is preferable to have the seating slightly low as chocks can be easily fitted to suit. Sufficient room under the motor should be allowed for a save-all to be fitted, and at the same time have sufficient room under the save-all, to allow bilges to be thoroughly cleaned out.

Motor Seating.—This should be carefully examined to see that it is securely fixed to the framing of the boat. Even for small motors, long fore and aft seating should be fitted and securely stayed athwartships, more especially if the motor is of the two-stroke type, as there is a greater tendency to vibration with this type of motor than there is with a four-stroke. Even for the smaller motors, fore and aft seating should always be fitted, not athwartships as is sometimes done, otherwise vibration soon becomes excessive and leads to hot bearings at aft end of motor and forward end of reverse gear if separate reverse gear is fitted. Should the reverse gear be embodied with the motor itself as is common on numerous types, trouble may arise in the gear box itself owing to vibration; cones will become jambed or fail to grip, bearings become overheated, gears wear rapidly, and even shafts break altogether at the point where one member is reduced in diameter and let into the other member to keep the shafts central and the cone faces in alignment. Where this occurs nothing short of lifting motor and reverse gear and fitting proper seating will rectify the trouble.

Many types of two-cylinder two-stroke motors have flywheels of fairly large diameter fitted at forward end of engine, which makes it impossible to run seating forward of the flywheel. When this is the case and it can be arranged, seating should be arranged (if made of channel iron, or plates and angles) to clear the flywheel; where seating is of wood, an additional member should be fitted, securely bolted through main seating with bolts placed horizontally, and also securely bolted to the framing of the boat. When bolts have to pass through a wooden hull for holding down purposes, they should have snap heads and large plate washers fitted outside. Plate washers need not be very thick, and should be let in flush with the wood, leaving bolt heads only projecting. The seating should be recessed to take nut and washer on the inside. When tightened up the end of bolt and nut should be under the flush of the seating. These bolts should pass through seating and framing only, and should not be utilised for holding down the motor. Separate bolts or stud bolts should be used for fixing the motor in place. There is no alternative however in some cases save to use holding down bolts for the motor with these passing right through seating and hull, and when this is necessary, bolts should be made of sufficient length, with snap heads screwed up far enough, so that when put in place and nut tightened down on to a washer in recess in seating by means of a tube spanner the bolt projects sufficiently to take the motor. When this is resorted to, great care must be taken to have holes marked and bored exactly to suit holes in bedplate of motor and keep it in correct alignment with propeller shaft.

Stern Tube.—When the line is in correct position through motor seating and hole in stern post or through stern brackets for twin-screw, reference marks placed vertically and horizontally 90° apart should now be placed on aft side of stern post equidistant from centre line, and also at a convenient place near position of forward end of stern tube. When this is done the line is removed, the arrangements are made to bore out stern post or brackets with suitable boring gear.

Having completed the boring out, the template taken of holes in flange at forward end of stern tube can now be applied to the bulkhead or framework at the forward end of stern tube, holes marked off and drilled, or this can be done if more convenient with stern tube in place. The stern tube being ready for securing, and fitting parts covered with read lead paint, it is drawn into place and tightened up with nut screwed home and securely fixed with a locking device. Wood packing may be fitted between flange at forward end of stern tube and bulkhead or framework and made secure.

The stern tube having been thoroughly cleaned out and lubricated with suitable grease, the propeller shaft is placed in position. In smaller motor boats the shaft can be pushed in from stern with propeller in place, the coupling having previously been removed. In larger sizes, especially where the motor is placed well aft, provision is made to draw the propeller shaft outwards, a special coupling being fitted on forward end to prevent the shaft being drawn out when running astern. In types where the propeller is working in an aperture, the propeller shaft must of course pass into the stern tube from the inside. For a right-handed propeller a left-hand screw should be cut on propeller shaft, and nut fitted, and *vice versa* when a left-handed propeller is fitted; yet with smaller installations this is not always observed. The stern tube stuffing box should be carefully packed with a suitable packing and well lubricated with graphite grease or tallow. After each ring of packing has been inserted the gland should be carefully screwed up until the box is almost full. The gland should now be screwed up fairly tight with a spanner, nuts then slackened back, set up hand tight and securely locked in place. Packing should be of suitable size for the stuffing box and ends neatly cut, so that they will butt fairly when pressure is applied on the gland. Should the coupling on the propeller shaft be removable the exact position should be noted, and with the propeller set to the proper distance from outer end of stern tube and marked from the face of coupling at forward end of propeller shaft to a fixed point inside the boat, with the motor in relative position.

Should the propeller be working in an aperture the clearance should also be noted between the tips of the propeller blades and the hull. With some types of friction reverse gears the propeller shaft is moved bodily either in a fore or aft direction when running ahead or astern. When reverse gear of this type is fitted, care must be taken that there is sufficient clearance between propeller boss and stern tube to allow friction cones in reverse gear to become properly engaged when running ahead.

Intermediate Shafting.—After the propeller shaft is settled the intermediate shafting (if any) with bearings, are placed in position and fixed temporarily, the final lining and setting up being done after boat is launched.

Sea Cock.—Considerable trouble often arises through a defective circulating water system, therefore a sea cock of ample bore should be fitted, so that should the boat ground, the inlet would not readily become choked. Where the boat is likely to run in shallow or muddy water two sea cocks may be fitted at different levels as a precaution, a grating in any case should be fitted to keep the passage to the cock as clear as may be. It is also an advantage to have a lip cast at the after end to hold the water at the opening when under way. It is advisable to have a mud box fitted with suitable strainer, jointed up to the sea cock as a precautionary measure against sediment reaching the circulating pumps and the cylinder jackets.

Exhaust Connection.—Should the exhaust pipe be led overboard near the stern, the connection should be fitted to the hull before launching, with a blank flange for the time being. The discharge connections should also be fitted in place and valves securely closed. With small installations, non-return valves are omitted; but where open connections are fitted for either bilge or circulating discharge, these require to be also safeguarded.

Fuel Tanks.—These, whether gravity or pressure fed, may be fitted either prior to launching or after, they are provided with suitable trays underneath to prevent oil getting into bilges. In the event of filling connections being let into the deck with water-tight cap, care must be taken to prevent water getting in. A vent pipe from the tank should be led outside to the atmosphere and so fitted that water cannot easily gain entrance. Fine gauze should also be securely fixed at the outlet to prevent any gas from igniting and reaching back into fuel tank should a naked light be inadvertently placed too near it. The gauge glasses fitted on the tank should have protecting guards. Test cocks are sometimes fitted, but the old method of dipping a rod is still common, while in some cases a gauge with indicator connected through simple mechanism to a cork float in the tank is fitted. The amount of petrol in the tank can be noted on the dial at sight. As this fitting is self-contained and screws into the tank, making an air-tight connection, it is more satisfactory than some of the methods already mentioned. All tanks should have suitable manhole or handhole doors for cleaning purposes; should tanks be too small to permit of this, large plugs should be screwed into the tank in a convenient position.

Reverse Gear.—Before reverse gear is put on board, should it be separate from the motor, suitable save-alls should be fitted. The gear should be carefully lined and set; the intermediate shaft having been previously lined up and securely bolted, the propeller and intermediate shaft couplings, also intermediate and reverse gear shaft couplings being tightened up and split pins inserted. With reverse gear in neutral position it should be quite easy to turn the whole shafting by hand or with a short lever.

Motor.—With several types of motors, provision is made to facilitate lining or setting up by means of adjusting screws through feet or flange on each side of bedplate. When the motor is lowered on to its seating, it is a simple matter to set shaft coupling to place by using the adjusting screws on the seating to bring the motor to the desired position. It can then be held by suitable grabs and the holes bored for holding down bolts. Chocks can now be fitted and the bolts secured. When a sufficient number of chocks have been fitted the adjusting screws may be withdrawn, the remainder of the chocks fitted, and the holding down bolts securely tightened up; the shaft couplings can then be tested finally before inserting the coupling bolts.

When the motor rests on wood bearers the same procedure for setting up the motor can be followed provided a small piece of plate is placed on top of bearer for taking points of adjusting screws, otherwise screws would sink into the wood. When the motor seating is of iron rather than wood, it is much easier to fit holding down bolts. In the latter case, stud bolts are usually employed. Where bolts or stud bolts cannot be used the next best thing is a suitable wood screw with head cut off, and the plain part or body screwed Whitworth thread. The wood screw is then inserted in the same way as an ordinary stud, and nuts used to tighten up. The crankshaft and reverse coupling bolts are now tightened up and split pins added. A little extra time is well spent when lining and setting up shafting, reverse gear and motor to get all in order and secure as it saves endless trouble afterwards. Should reverse gear and motor be combined, procedure would be as already set forth, except that they would be set up together.

Exhaust Pipe.—This can now be fitted, and in the majority of petrol and petrol-paraffin installations the pipe is led out near the stern above the water line or through the side, above or under water, and sometimes where a funnel is fitted it may be led up through a silencer into the funnel. In this case silencers are water-cooled and circulating water is then discharged direct overboard and not into the exhaust pipe as is usual when the exhaust is led out as previously mentioned. When designing the exhaust pipe it should be noted that the bore must at least equal the bore of the outlet on exhaust silencer, or manifold, if a little larger, so much the better. The pipe should be led with as few bends as possible; where bends are necessary they should be easy, and sharp bends should always be avoided. A silencer, sometimes water jacketed, is fitted in a convenient position between the exhaust manifold on the motor and the exhaust pipe outlet. When it is intended to discharge circulating water through an exhaust pipe, bends where water may collect should be avoided, otherwise, should motor chance to "kick" when starting up, and run a few revolutions in the wrong direction, water may be drawn into the cylinders. When an underwater exhaust is fitted it is necessary to have a small hole drilled, either in the exhaust manifold itself or somewhere near the highest point in the exhaust pipe so that air may enter as soon as engine is stopped and prevent a partial vacuum being formed in the exhaust pipe, and manifold, thereby tending to draw water up into the manifold and cylinders. In the larger sizes, so fitted, I think it is better to fit, say a $\frac{1}{4}''$ bore pipe from the exhaust pipe near its highest point, and lead same direct overboard above the water line. A small cock on the exhaust pipe would suit, but it is always liable to be left shut when it should be open, whereas the small pipe led outside acts automatically.

PRACTICAL NOTES ON PETROL, PETROL-PARAFFIN & SEMI-DIESEL ENGINES.

Water Circulation.—The suction pipe should be led from the sea cock or mud box with as few bends as possible, to the circulating pump. Bore of pipe should be at least equal to the bore of the connection leading into the pump. if of the geared type, or the connection on the valve chest, if of the reciprocating type. Care should be taken to see that all pipes and connections on the discharge side of the pump have at least the same area as the connection itself on the pump, or preferably larger to facilitate discharging conditions. With motors of the petrol type, circulating water passes through cylinder jacket, thence through cylinder cover (should motor be of the type fitted with cylinder covers), then through water jacket on exhaust manifold on motor and through the water jacket on exhaust silencer, when so fitted, or it may be taken direct from the exhaust manifold jacket and led into the exhaust pipe itself as near the exhaust manifold as convenient for fitting; the connection should be just sufficiently far below the exhaust manifold that circulating water will not run of itself or be drawn into the exhaust manifold or cylinders should the motor kick back when starting up, and probably run a few revolutions in the reverse direction. When petrol-paraffin motors are fitted the exhaust manifold as a rule is not water cooled, but covered with asbestos and planished steel, as the vapouriser is usually embodied in the exhaust manifold casting. When gear or drum pumps are fitted no water would be discharged through jackets should motor kick back when starting up, but where a plunger pump is fitted, water is very liable to get into the cylinders through exhaust valves; this sometimes causes considerable trouble. On quite a number of marine motors the circulating discharge is often found on the cylinder heads placed on the athwartship centre line of each cylinder or pair of cylinders if cast in pairs. Provided the motor is installed with little or no rake nothing serious will happen, but in the majority of cases motors, for various reasons, especially if the line of shafting is short, must have a certain amount of rake. This rake causes pockets to be formed inside the jacket at the forward end of the cylinder head. which very often leads to cracked jackets. The circulating discharge should in all cases be taken from the jacket at the forward end of the cylinder head.

Circulating Pumps, as used on marine motors, are, generally speaking, of three distinct types: plunger, drum, and gear pumps. Plunger pumps are usually found on the slower running motors, and are either driven direct off the cam shaft or through reduction gearing. They are usually of the single acting type, although double acting pumps are occasionally to be found. When a double acting pump is fitted, one side of the pump is used as a circulating pump, while the other side is used as a bilge pump. This, I consider, is bad practice. Sooner or later, owing to the wear of the pump plunger rings or packing, water finds its way past the plunger from the circulating side to the bilge side and *vice versa.* Circulating water passing out through the bilge discharge does no harm, except in the loss of water, but bilge water, which practically always contains a fairly large percentage of oily matter, finds its way into the cylinder jackets, and the inevitable result is, cracked jackets, through it becoming deposited on the hot walls of the jacket, thereby destroying the efficiency of the cooling water. When a single acting plunger pump is fitted it gives the best service. It will run for long periods without requiring overhaul, and even when working under the worst possible conditions it is not a costly matter to overhaul it.

Drum Pumps.—These pumps, which are of the rotary type, consist of a rotor with a slot cut through it longitudinally, into which two vanes are fitted. These vanes are pressed outwards by means of phosphor bronze springs. The rotor is then placed within its casing, the end covers (with necessary stuffing box) are bored eccentric to the bore of the casing itself. As the rotor revolves the water is swept out and discharged by means of the vanes fitted within the rotor. Pumps of this type stand up to their work for a reasonable time, provided they are not working in gritty water. When working in gritty water casing and vanes soon begin to wear, and a pump of this type is not so easily or so cheaply overhauled as one of the plunger type.

Gear Pumps.—Gear pumps, which are found on a large number of motors, give endless trouble through the same cause as stated with drum pumps. With gear pumps, casings, spindles, and pinion teeth wear rapidly when working in gritty water, and the only remedy when wear begins to assert itself is to fit a new pump, which will very probably save cracking of cylinder or exhaust manifold jackets. These pumps as a rule run at engine speed, therefore their life is usually very short, especially when working under the above-mentioned conditions. Both drum and gear pumps, especially the later type, when circulating water is being discharged into the exhaust pipe, soon show signs of trouble, through cylinder jackets becoming overheated, as soon as wear begins to assert itself in the pump. Unless the pump can discharge the water into the exhaust pipe against the pressure of the exhaust gases this pressure asserts itself on the pump, with the result that water is churned in the pump casing, and overheating of the cylinder jackets immediately follows.

With all motor boats, as soon as the motor has been shut down, all oil and water connections should be closed at once. This is specially the case where the motor has been fitted with a drum or gear type circulating pump, as the cylinder jackets will immediately commence to drain out through the pump as soon as the motor stops, resulting in the jackets being empty when starting up, and overheating very likely to occur, unless care is taken to fill up the jackets, which is not always a very convenient thing to do.

All cylinder jackets should be provided with suitable cleaning doors or plugs, easily accessible, to enable cylinder jackets to be thoroughly cleaned out at regular intervals. When cleaning doors or plugs are difficult to get at, cleaning out of the jackets is usually neglected, with serious results.

Where circulating water is being led direct overboard, after passing through cylinder and exhaust manifold jackets, the outlet, where possible, should be sufficiently high to ensure a small head of water in the jackets. A non-return valve is sometimes fitted to gain this object.

All motors intended for service in tropical climates should have larger circulating pumps and connections fitted than those intended for service in home waters, otherwise cylinder water jackets will run extremely hot, which will very probably cause pre-ignition, with its attendant troubles.

Fuel Supply System, as fitted on petrol and petrol-paraffin installations, is either gravity or pressure fed. Gravity system is commonly used in small powered motor boats. Petrol tank or tanks are usually placed in position about 2ft. or even less above the carburetter, just sufficient to give the necessary head of petrol on the needle valve of the carburetter. On some of the larger petrol-paraffin installations, where a much larger supply of fuel must be carried, main fuel tanks are fitted in the most convenient part of the boat, from which it is pumped into a small gravity supply tank by means of a small hand operated semi-rotary pump. In some instances a small geared pump driven off the motor supplies the gravity tank, an overflow pipe being fitted from the supply tank, back to the main tank. A small hand operated semi-rotary pump is usually fitted as a stand by. When gravity tanks are fitted they must be properly ventilated to the atmosphere.

Pressure Fed System.—The fuel tank is fitted in the most convenient place in the boat. Petrol or paraffin is forced to the carburetter by means of compressed air, or exhaust gas from the motor itself, at a pressure of 2 to 3 lbs. per square inch, which is admitted at the top of the fuel tank. A cap or plug fitted on the filling pipe of a pressure fed tank must be carefully jointed to prevent loss of pressure. A hand air pump is fitted to obtain the initial pressure for starting up. When compressed air is used a small air compressor is driven off some convenient part of the mechanism. This air compressor which is simply a plunger working in a small cylinder about 1″ or 1¼″ diameter, uncovers several little holes or ports when at the end of its outward stroke. On the inward stroke, air is discharged through a small spring loaded non-return valve fitted on the side of cylinder exactly at the end of the plunger's stroke, from which it is led to the fuel tank. To regulate the pressure, the air compressor cylinder cover takes the form of a long screwed plug, screwed into the cylinder, above the non-return valve and held in place by means of a lock nut. Pressure can then be increased by screwing the plug further in, thereby reducing the clearance between plug and pump plunger, or *vice versa* to reduce pressure as required. As the screwed part in top end of cylinder stops against a shoulder it is impossible to screw the plug too far in and foul the pump plunger.

Exhaust gases tapped off exhaust manifold is a method very commonly used. When petrol or paraffin requires to be pressure fed a special fitting containing a non-return valve is fitted to the exhaust manifold. This non-return valve is fitted with an external spring, and a screw for regulating the valve lift, or closing the valve off altogether if necessary. It requires to be examined at frequent intervals, as the spring is liable at times to get overheated, and become weak or useless altogether. The pipe between valve and fuel tank requires to be examined occasionally, as it is liable to become choked at the bends through particles of carbon being carried over with exhaust gases. A small hand air pump is also necessary to obtain initial pressure in fuel tank before starting up.

PRACTICAL NOTES ON PETROL, PETROL-PARAFFIN & SEMI-DIESEL ENGINES.

A suitable filter should always be fitted in the pipe line between the tank and the carburetter in a position where it is easily accessible for cleaning purposes. Piping between tank and filter, and filter and carburetter, should be free from unnecessary bends, to prevent air locks forming in the pipes, which are soon demonstrated through irregular running, or the actual stoppage of the motor, through temporary lack of fuel. All piping throughout the boat should be properly clipped up and fastened to place, to prevent joints becoming broken through vibration.

Carburetters.—There are various types of carburetters on the market, and motor manufacturers either fit their own make or one of another make, which they have thoroughly tested on their motors, and found it gave the best results under all conditions of load. There is nothing much to get out of order in a carburetter provided it is not tampered with, and provided every precaution is taken to prevent grit from getting in and choking the jets. The float may leak (metal floats are usually fitted) and when this happens the carburetter will commence to overflow. When this is seen the float with its needle valve should be taken out and the needle valve examined to see that it seats properly. Should the trouble be with the float itself, that can be easily ascertained by gently shaking the float. Should there be a leak, it may be necessary to drill a small hole to get the petrol or paraffin out, then the hole can be tinned over with a soldering bolt, also any other part that may look suspicious. Many types of carburetters have their induction pipes connected to a metal casing, placed around an uncooled part of the exhaust pipe or the exhaust manifold itself. This is done to heat the incoming air, thereby causing better carburretion of the mixture to the cylinders, and facilitates starting up, especially during cold weather. A connection is sometimes fitted between crank case and carburetter, whereby the vapour is drawn from the crank case and into the cylinder through the carburetter.

Lubrication.—There are two systems of lubricating petrol or paraffin motors, splash and forced lubrication. With splash lubrication the crank case is filled up until the heads of the bottom end bolts just dip when crank is on the bottom centre. An oil box is fitted either on a dashboard or the cylinder jacket, from which the lubricating oil is led to the crank case, and probably the forward main bearing, through sight feed lubricator which can be regulated at will. When the motor is running, oil is thrown up, and lubricates gudgeon pins, cylinder walls, main bearings and bottom ends. A float with a rod attached passing out through the crank case is sometimes fitted, to show the amount of oil in crank case, the rod being marked to show proper oil level in crank case. Provided the line of shafting is almost parallel with the water line no serious trouble may be anticipated, but as is usually the case the motor is fitted at more or less of an angle to the water line, consequently the lubricating oil in crank case flows towards aft end, the result being the forward bottom end is running dry, while the aft bottom end is partly submerged. Various devices have been tried to trap the oil as it drips down and lead it back into the well under the forward bottom end, but they are seldom satisfactory where the motor has been installed with too much angle. That is one reason why the forward main bearing is supplied with a separate feed from the lubricating oil box, on the dash board or cylinder, to ensure it being properly lubricated. Another trouble that often crops up in addition to forward bottom end troubles, through lack of lubrication, is over lubrication in the aft cylinders. An excessive amount of oil is thrown up on to the cylinder walls, especially the aft cylinder. Oil gets up past the piston, causes smoky exhaust and dirty sparking plugs, which short circuit and miss-fire, causing irregular ignition; all this leaves the combustion chambers, pistons, and exhaust valves in a filthy condition.

Forced Lubrication.—When forced lubrication is fitted, the angle the motor is installed at is not of so much importance. A well is cast in the bottom of crank case to receive the lubricating oil. It is drawn from this well through a strainer by means of a pump. A gear pump is often fitted for this purpose, and discharges at a pressure of from 10 to 12 lbs. per square inch, to main bearings and bottom ends, the oil thrown off the moving parts being sufficient to lubricate the gudgeon pins and cylinder walls. This method is preferable to splash lubrication

Ignition.—High tension ignition is now chiefly fitted on all petrol and paraffin motors, although there are still a number of the slower running type fitted with low tension ignition. The larger types and even a good number of the smaller types of petrol and petrol-paraffin motors are usually fitted with a battery and induction coil to facilitate starting up. When this system is employed, two sparking plugs per cylinder are fitted, with separate leads, *i.e.*, from magneto to sparking plug, and from battery through induction coil to the other sparking plug in the cylinder head. The motor is started up, off the battery, and then switched over on magneto, the battery only being used for starting purposes.

Magnetos.—If kept thoroughly clean, free from dust and moisture, and not over lubricated, will run for a considerable time without giving trouble. They should, however, be examined periodically, to ensure their being kept in good running order. On these occasions platinum points on contact breaker should be examined, and if necessary, filed absolutely flat with a very small flat file, and finished off with fine emery cloth. When finished the faces of the platinum points should make absolute contact, to prevent, as far as possible, sparking taking place when contact is broken, as excessive sparking very soon destroys the platinum points. The gap between platinum points when contact is broken, should be set to gauge, usually supplied by the makers. Should no gauge be available, the gap should be approximately one 1/32 in. Distributor should be carefully examined, and all oil or carbon deposit carefully wiped off the commutator. Faces of brushes should be carefully cleaned of any deposit. As the face of the carbon brush will have worn itself to fit exactly on the commutator, it should be cleaned with a rag soaked in petrol. It is not advisable (unless absolutely necessary) to file the face of the brush, but when this is done it should be carefully bedded to the commutator, by inserting a strip of emery cloth around the commutator, with emery side towards the carbon brush. By this means a fairly good surface will be obtained on the face of the brush. All emery or carbon dust should be carefully cleaned off. the various parts assembled, and the magneto should again be in good working order. When these points have been seen to, magneto should be tested, to ascertain if a good spark is available at the sparking plug, this can be ascertained by withdrawing sparking plugs from cylinder heads, earthing plugs themselves on top of cylinder or other metal part of the motor and turning motor rapidly by hand, when spark can be easily seen at the points of the sparking plug. Everything else being in order, if it is found impossible to obtain a spark in this manner, magneto should be handed over to the makers, or a reliable firm, for repairs. On the larger installations the same test can be carried out by disconnecting magneto, and revolving armature by hand, care being taken when replacing magneto, to put it back to its proper setting. A switch is usually fitted, so that the motor can be stopped, by simply earthing the current direct from the magneto. It is as well to remember that when testing magneto as described, this switch should be open, otherwise it will involve an unnecessary waste of labour.

Methods of Starting.—The usual method of starting up a motor is either by hand, by electric self-starter, or by compressed air. The method of starting up motors by hand is well known, so that nothing further need be added. Electric self-starters are fitted on a large number of American built motors, and given reasonable care when being operated, will be found very satisfactory. The starting motor, fitted on to the motor itself, has a small pinion on the armature shaft which engages with suitable teeth cut in the rim of the flywheel when the current is switched on to the motor only. A fairly large battery is necessary to operate the starting motor, and on most installations a small generator is fitted, also driven off the main motor, which charges up the battery while motor is running. A foot pedal switch control is usually fitted. When starting up, the pedal is pressed down on to the first contact, which magnetically draws the armature of the starting motor on and against a spring on the armature shaft, thereby bringing the pinion into gear with the teeth in the rim of the flywheel. The pedal is then pressed full down when the electric motor will commence to drive the motor. When the petrol motor commences firing, the foot pedal is released, current is cut off, and the spring already mentioned presses armature back to its original position, at the same time withdrawing pinion on the armature shaft out of gear. The engaging ends of the teeth on the flywheel and starting motor pinion are filed V shaped to facilitate meshing of the gear. Care must be taken when pressing down pedal control for starting, not to press down too rapidly, otherwise ends of the teeth may be badly chewed up, owing to starting motor revolving rapidly, before gears are properly in mesh.

Electric generators driven off the motor itself are often found driven through a free wheel device to prevent armature being driven in the reverse direction should the motor kick when starting up and run a few revolutions in the wrong direction. A friction driving device is sometimes fitted to prevent armature being driven at excessive speed should motor be allowed to race, the friction device slips when the speed reaches a predetermined limit. With six cylinder motors, these are very often started up by simply turning on the petrol supply, adjusting the throttle and after priming cylinders, cranking round the motor two or three revolutions, then switching on the current from the battery through the induction coil, when motor should start. So long as the pistons and rings can retain a fairly good compression in the cylinders, motor may be stopped for a short period, and started up again by simply switching on the current. Compressed air is occasionally used for starting up the larger petrol or paraffin motors, and when it is so a mechanically operated distributor is fitted on the motor to admit air to the various cylinders at the proper time, this is immediately cut out as soon as the motor commences firing.

Before starting up a motor for the first time after it has been installed, the motor should be thoroughly examined all round, to make sure everything is in order. Crank case should be thoroughly clean, and sufficient lubricating oil in crank case, or well, according as splash or forced lubrication is used. Gear box should be attended to in a like manner, and all grease cups on intermediate shaft bearings and stern tube filled. Carburetter is now tested to ensure that an ample supply of petrol or paraffin is available. Sparking plugs tested to ensure all are in good working order. After being replaced in cylinder cover or head, sea cock should be opened, throttle on carburetter should be partially opened, magneto retarded, and magneto switch in its proper position, or, if starting up off an accumulator and induction coil, switch should be in. Care should always be taken to see that ignition is retarded before commencing to crank round the motor, otherwise motor might kick back when starting up, due to ignition taking place before the crank was over the dead centre. As soon as the motor starts up, switch off accumulator and run on magneto only.

See that lubricating oil and circulating water supply is satisfactory. Motor should be run light, for a short period, to see that everything is in order. The propeller should be put in gear, the boat having been properly moored up previously, and the motor given a run under load. Having run for some time ahead, it should then be tried astern. Motor should always be throttled down before commencing to operate reverse gear. After having run for a reasonable time motor should be stopped, crank case doors taken off, and all bearings felt round. Gear box should be examined in a similar way, then, assuming everything has been found in order, crank case doors can be replaced and the boat proceed on trial.

PRACTICAL NOTES ON PETROL, PETROL-PARAFFIN & SEMI-DIESEL ENGINES.

Semi-Diesel Engines.—My experience with semi-Diesel engines has been chiefly with the two-cycle type, fitted with either solid or air injection. The installation generally is carried out on the same lines as on the smaller petrol or paraffin engines. Care must be taken to ensure that the engine is rigidly bolted down to its seating, that the seating is of ample length, and securely fixed to place. The fuel tanks, usually of fairly large capacity, are fitted in whatever may be the most convenient part of the engine room, filling pipes being fitted to water tight deck connections, and vent pipes led through engine room casing or other convenient part, to the atmosphere. The fuel oil is usually pumped by means of a small hand semi-rotary pump to a gravity tank, fixed on the bulkhead or other convenient place above the level of the fuel pumps on the engine. On the larger installations a service pump is fitted, which keeps the gravity tank always full. This pump is fitted to the main engine and runs continuously, an overflow pipe being fitted near the top of the gravity tank, which connects with the main fuel tanks. Gravity tanks should not be placed too high above fuel pumps, otherwise oil may pass through fuel pumps into cylinders when engine is at rest and lead to serious consequences.

Solid Injection Type.—The fuel pumps are of the plunger type, a separate pump being fitted for each cylinder, and they are usually operated through a "hit and miss" governor,, although a number are controlled by means of a centrifugal governor, which controls the travel of the pump plunger by means of a wedge piece inserted between the mechanism operated by the cam itself and the plunger. This wedge piece is either inserted or withdrawn by the governor gear, thereby increasing or decreasing the travel of the pump plunger. Another method of governing is by moving a cone shaped sleeve which carries the small cams for operating the fuel pumps, endwise along the shaft, thereby increasing or decreasing travel of fuel pump plunger. With the centrifugal governor the amount of oil injected into the cylinder each revolution is increased or decreased as the speed of the engine demands. The discharge connection of the pump is connected direct to an injector, which is fitted into the cylinder or cylinder cover, through which the oil is injected into the bulb, previously heated externally by means of a blow lamp. It is essential that a non-return valve be fitted as near the nozzle of the injector itself as it is possible to get it, to prevent the gases from getting back into the pipe or even back into the pump. Owing to the stroke of the pump being very short, the smallest amount of gas or air pressure in the pipe or the pump will put the pump out of action, thereby cutting off the power from that particular cylinder, with consequent reduction in speed or stoppage of the engine altogether. It is therefore necessary to avoid this, and discharge and non-return valves should be regularly inspected and kept in order. There are various types of injectors to be found, each maker favouring his own particular type. some types spray the oil into the combustion chamber more or less broken up, others again, inject it in more or less in a solid stream until it impinges against the hot walls of the combustion chamber, where it is immediately vapourised and ignited. Whatever type is used it is important that the amount of oil as measured by the travel of the fuel pump plunger should be instantaneously cut off, when piston reaches end of its stroke, and not allowed to slowly trickle through the nozzle of the injector when piston is descending on the power stroke. Should this happen it will cause dirty pistons, and a smoky exhaust. This trouble is mainly due to gases leaking past the non-return valve, into fuel discharge pipe, which for a time acts in a similar manner as an air vessel on a pump. Once a certain amount of gas has got into discharge pipe no fuel will be injected at all, and the engine will cease working on that particular cylinder or the engine may stop.

Lubrication.—Lubrication of the two-stroke engine presents more difficulties that it does with engines of the four-stroke type, owing to the enclosed crank case, which is necessary for scavenging purposes. It is essential, for two reasons, that the crank case be made as air tight as possible, first to obtain the highest pressure of air it is possible to get for crank case compression for scavenging purposes, and secondly, to prevent air from blowing the lubricating oil out of the main bearings. The usual method of preventing air getting into main bearings is to fit sealing rings, fitted together in halves, on the crank shaft journals. These rings are made a good fit on the journal, and are kept in position against the machined face of the crank case by means of springs fitted into recesses drilled into the crank web, and also into the sealing ring. Two driving pins securely fixed into sealing ring, project into clearance holes drilled in the crank web, making the ring positively driven. The springs require to be just strong enough to keep the ring up against the machined face of the crank case. Main bearings can then be quite easily lubricated by means of drip feed lubricator or the ordinary oil box, fitted with syphons. In some instances ring lubricators are fitted, but these are not common. In other cases grease is employed, fed in by means of spring fed grease cups. Lubricators, positively driven, of the sight feed and pressure type, are principally fitted for lubricating bottom ends, top ends, and piston and cylinder walls, each connection being fitted with a hand regulating device to give more or less oil, as required.

It is very important to keep all bearings properly lubricated without wasting lubricating oil, and of the three bearings, the crank-pin usually gives more trouble than any other. This bearing is, as a rule, fitted with a centrifugal oiler securely fixed to crank web, through which a hole is drilled, leading out on to the crank-pin. The oil is led into the centrifugal oiler from the lubricator by means of a connection passing through the crank case, and securely fixed to it. As the amount of room in the crank case of engines of this type is limited, it is exceedingly difficult to lead a lubricating oil pipe from connection passing through crank case to near the bottom of centrifugal oiler. In addition, this pipe has to be securely fastened in place to ensure the oil always dripping into the centrifugal oiler. It has been found very satisfactory to keep the groove in the centrifugal oiler into which the oil is caught when it leaves the pipe reasonably broad, and by so doing, a straight connection screwed through the crank case as near the horizontal centre line of the shaft, or lower if it can be conveniently placed, has been found very satisfactory, especially if the connection is made with a circular beading turned on the end of it, to prevent the oil from creeping back along the connection, and dripping off into the crank case past the centrifugal oiler. Provided bearings and lubricator connections are properly fitted very little trouble may be expected from bottom ends. Lubrication of top end bearings is accomplished in the majority of engines by means of a connection passing through the water jacket and cylinder wall to the cylinder (care being taken that this connection is made absolutely water tight, otherwise water will leak from the jacket into the cylinder and crank case). This connection registers with a scraper which is fitted into the hollow gudgeon pin, and kept up against the cylinder wall by means of a spring fitted behind the scraper inside the pin. The oil passes through this scraper and then through holes drilled in the gudgeon pin itself, which leads the oil into oil grooves cut in the top end bushes, especially the bottom half, from which it is distributed over the surface of the bearing. The piston is lubricated by means of similar connections carried through the jacket and cylinder walls, placed at regular intervals around the cylinder. In the smaller size engines two connections only are used for lubricating the piston, in addition to the top end lubricating connection, which also assists in lubricating the pistons. In some cases lubricating oil is carried up to the top end through a hole drilled in the connecting rod, oil from the bottom end passing up through a small non-return valve, fitted inside the oil hole at the butt end of the connecting rod after it has lubricated the crank pit, but this method is not very common. It is necessary to keep crank case of two-cycle engines well drained, so as to prevent any lubricating oil that gathers from being thrown up by the bottom end into the cylinder aided by the scavenging air, and thus causing the engine to knock heavily. Drain pipes with cocks are usually fitted to each crank case, and these should be drained at regular intervals. It is also well to bear in mind, once the engine has been run, never to take off crank case doors and attempt to look inside with a naked light, lest there may be an accumulation of gas.

Water Circulation.—On the majority of two-stroke engines a single acting plunger pump is fitted, worked off an eccentric which operates the fuel pump gear. On an engine with two or more cylinders, each of which is a separate unit, it is advisable that a valve or cock should be fitted on each cylinder where circulating water enters the cylinder jacket. A relief valve should always be fitted on the discharge side of the pump, in case these valves or cocks should be left closed when the engine is started up. The object of fitting means to regulate the water supply to each cylinder is to control the water supply should one or more cylinder jackets become excessively hot while others remain cold. This may arise from one or more causes; in the first place it must be remembered each cylinder has its own fuel pump, and it may happen that one cylinder is getting more oil delivered to it on each stroke of the fuel pump than its neighbours; consequently it is getting too much oil and the excessive heat generated in the cylinder, thence to the jacket, tends to prevent the entrance of the circulating water, which naturally seeks the path of least resistance via the cooler jackets. To remedy this the stroke of the fuel pump which supplies this particular cylinder should be immediately reduced, and the water inlet cocks or valves on the jacket regulated so as to ensure the temperature in the hot cylinder being reduced before the temperature reaches that stage where the walls of the cylinder cover jacket crack, through unequal expansion between inner and outer walls of the cylinder jacket. On an engine where there are numerous bends on the circulating discharge pipes this trouble is more liable to happen than would otherwise be the case if the pipes were more free from bends. Owing to the steam generated in the jacket with excessive heat, locks occur in the bends which eventually may cause damage, unless means are provided to regulate the temperature on each cylinder.

Bilge Pumps.—Bilge pumps of the single acting plunger type are usually fitted. They are driven off the eccentric operating the fuel pumps, or through a separate eccentric; should the fuel pumps be operated off the governor shaft, bilge suction pipes are usually led into strainer boxes placed in the bilge.

Starting Up.—When the engine is directly reversible by means of compressed air no clutch is used at all. Starting up either ahead or astern is affected by means of a hand wheel on the engine, which admits air through valves, sometimes of the piston type, to the various cylinders, in proper sequence. These piston valves are operated off the fuel pump mechanism, and are cut out as soon as the starting air is cut off, coming automatically into action whenever the starting air is turned on. They are not in use when the engine is running on oil. With the exception of the very smallest sizes, external means are always employed for starting up, compressed air or exhaust gas, tapped off from the cylinder itself, when ignition takes place, through a special non-return valve, which is also utilised as a starting valve, for starting up. When the installation does not include an auxiliary air compressor, two or three air bottles are usually fitted, connected with necessary valves and piping to the starting valve on the cylinder cover or covers as the case may be. One of the bottles, usually smaller than the others, is connected to a hand operated air compressor by means of which the initial start is obtained. When the engine is running, exhaust gas is tapped off the cylinder in which the starting valve is fitted, into the air bottles, to a pressure slightly below the initial pressure of the explosion in the cylinder. When starting up either two or four cylinder engines, the engines must be turned round until the crank is just over the top centre. the starting valve being fitted on this particular cylinder. Engines started up in this manner are fitted with friction clutch or reverse gear kept in the neutral position when starting up. Bulbs are then heated by means of blow lamps, and when sufficiently hot, fuel pumps are given one or two strokes by hand to ensure that they are in working order, and to inject the initial charge of oil into the combustion chamber. A valve or air bottle is now opened, and the engine is given an impulse by means of air or exhaust gas, admitted through starting valve. Assuming everything is in order the engine will immediately start up. Should the engine be running free, with reverse gear in neutral position, the pressure may not rise very high in the starting air bottles, owing to the engine running light, but will immediately go up when the load is put on.

Air Injection Type.—This type of engine requires more gear about it than the solid injection type. The special design I have in view is, in operation, exactly similar to the Diesel engine, excepting as to the initial compression, which is very much lower than that generally used, consequently a blow lamp is necessary for heating up. Engines of this type require a two-stage air compressor with intercoolers, and as they are directly reversible by means of compressed air, fairly large air receivers are necessary. An auxiliary air compressor, driven with a small petrol-paraffin set, is fitted. This auxiliary set, in addition to driving the compressor, is sometimes used for driving the lighting set and auxiliary circulating and bilge pumps, through suitable gearing and connections. There are not many engines of this type in use, as the solid injection type, being much more simple and easier to construct and run, is usually installed.

Starting up ahead or astern is effected through a hand wheel operating a small cam shaft, which admits starting air through a mechanically operated distributor to the various cylinders, and at the same time brings the fuel pumps into action. As soon as the engine picks up on oil the starting air is immediately cut off.

When engines of either the solid or air injection type are running, a fair idea may be gained if each cylinder is doing its own share of the work, by noting the condition of the respective bulbs which should show dull red when running at normal load; with one dull red, and another a bright red, it is evident one is getting more fuel than the other, and the stroke of fuel pumps should be adjusted accordingly. Should this not have the desired effect the injector should be examined to see that it is not choked, or the discharge valve may not be seating properly in the fuel pump itself.

When engines, irrespective of type, are stopped even for a short period, all cocks on fuel tanks should be shut off, thereby preventing any risk of petrol or oil leaking into the bilge, or even into the cylinders. Sea cocks should also be shut to prevent water draining out of the cylinder jackets, more especially when gear pumps are fitted. These are necessary precautions that should always be taken, and when attended to, the engineer saves, ultimately, time and trouble.

The CHAIRMAN: Col. Lamb has till recently been responsible for the upkeep of about 600 engines in motor craft on the rivers in Mesopotamia, this giving an experience in upkeep which should cover most of the defects in running and maintenance of interest to engineers.

A MEMBER: I think the Author's classification is somewhat indefinite. He commences by dividing the paper into three sections: 1, Petrol; 2, Petrol Paraffin; 3, Semi-Diesel, Air Injection and Solid Injection. He, however, leaves it at that, hence throughout the paper we are not quite sure in many instances what particular type of engine he is referring to. He states that in the two-stroke engine there is a greater tendency to vibration and further on he quotes that the installation of a two-stroke semi-Diesel type is generally carried out on the same lines as for small petrol and paraffin engines. Such a statement clearly requires further qualification, not only with regard to types, but also size.

He speaks of a special design of air injection engine exactly similar to the Diesel engine, except as to the initial compression which is very much lower and therefore a blow-lamp is used. If it is a Diesel it does not require a blow-lamp, and if a blow-lamp becomes necessary it is not a Diesel. Then, as to the term "solid injection," I prefer "mechanical injection."

As regards the question of installation, I think that the Author might have conveniently shown a few simple diagrams indicating the general lines on which the foundations should be constructed. Take the particular instance of wooden ships, they are being converted continuously, and a few hints as to the method to be adopted would have been very useful. If you make the foundation too stiff you are probably straining the hull, and if you are not making it strong enough you are probably straining the engine.

Then there is the question of dock trials. Full load mooring trials probably overload the engine, and if the adjustments are made under these conditions there is a possibility of producing conditions which are not conducive to best results at sea. A few hints on this point would have been very useful.

The Author refers to vapourisers, but does not give his experience as to which is the best type to be found in practice. It is an important subject because people are apt to discuss the waste of oil fuel too readily without being familiar with the facts as they are to-day. It is said that oil-fired boilers are wasteful and that they should be immediately replaced by internal combustion engines. It should, however, be remembered that the oil which is to-day looked upon as an excellent fuel for oil-fired installations cannot be used with the same degree of reliability and overall efficiency in an oil engine. A better grade of fuel would therefore have to be used resulting in a great amount of heavy residual oil losing its commercial value. On the other hand, with regard to petrol and kerosene engines there are a number of engines which can use kerosene satisfactorily. If, therefore, petrol engines are installed it is decidedly a question of wasting fuel.

The Author makes rather an interesting statement when he talks about the "air injection semi-Diesel engine." I should be interested to know what particular type he is referring to as in my own experience hot-bulb engines (which I take it he refers to as being semi-Diesel engines) generally operate on the two-stroke principle and have mechanical injection. I do not think he refers to the Bolinder type because compressed air is not used to inject the fuel but is mixed with it at the nozzle orifice and is said to have a cooling effect on the combustion chamber. He also states that the general type of hot-bulb engine is of the "hit and miss" type governor. In this country this is certainly not the case and the standard practice for marine hot-bulb engines is an ordinary pump feed where the feed pump injects the charge proportionately to the load. Again the utilisation of exhaust gases, by-passing them into a reservoir for starting up purposes is in my opinion not sound. Conditions may arise beyond the control of the engineer-in-charge with a result detrimental to the plant. Comparatively small H.P.'s can be readily started by hand, and there are on the market to-day small hand compressors which are quite sufficient for all practical requirements. In many instances the starting arrangement is combined with the pumping up valve; gumming up of that valve is largely attributable to this method of re-charging and starting.

The Author: The Member spoke first of all of the vibration in the vertical two and four-stroke engines. If I mentioned that there was less vibration with the two-stroke, I made a mistake. There is always more vibration with the two-stroke than with the four-stroke. I have always found that to be my experience, that is to say, more vibration with the two-stroke than with the four-stroke.

Member: This is just where we are not quite certain which type of engine you are speaking of. I do not think it fair to compare a slow speed two-stroke vertical hot-bulb engine with a high-speed multi-cylinder petrol or paraffin engine.

The Author: I have only seen a few vertical four-stroke hot bulb marine oil engines, and these were chiefly Continental. There was more vibration with the two-stroke than with the four-stroke engine. Speaking of magnetos, keep them clean and dry. It is not always possible to keep them so in barges and coasters, but the necessary care should be taken. If kept clean and dry they will serve you pretty well provided they are not interfered with. The member also spoke about semi-Diesel and solid injection engines as being mechanical injection. I believe he is right in describing the solid injection type as mechanical injection, although it is generally known as solid injection, hence my reason for using the term. The semi-Diesel engine proper is an engine that uses compressed air for injecting the fuel into the combustion chamber. Not many of these are being built to-day to be fitted into vessels, but Messrs. Blackstone, I believe, are still doing it. About nine years ago I was round here on the Thames with a boat fitted with a 120 B.H.P. semi-Diesel engine. This engine was fitted with a direct driven two stage air compressor which supplied injection and starting air, compressed air at 400 lbs. pressure being used for injecting the oil into the combustion chambers. That I should call a true semi-Diesel oil engine. A number of the members of this Institute at that time were given a demonstration trip on the Thames. Speaking of dock trials and overload you will always get a little overload with dock trials, yet I am not prepared to say how much you will get exactly, but it has always been my practice to have a good dock trial before letting the ropes go. There are various types of vaporisers made by different manufacturers, and they all more or less give a certain amount of trouble, and I do not think any one is so much superior to its neighbours. The member also referred to the use of heavy oil in Diesels instead of using it as fuel in furnaces, but I am not quite clear what he meant. The heavier fuel oils are not burnt in semi-Diesel engines, but they can be easily burnt in the Diesel engine itself. With a Diesel engine which burns very heavy oil they usually use a lighter oil when starting up, and heating arrangements are provided, whereby the heavy oil is heated by means of the exhaust gases before being used. After an engine has been running on this grade of oil it would be necessary to switch on to the lighter oil before shutting down the engine so that the engine would be ready to start up immediately it was required.

Member: My point is this. People state that we are wasting fuel, of which there is only a limited amount available, by using it in boilers where it can be burnt less efficiently than in internal combustion engines. I should be the first one to advocate the installation of the oil engine, but we have a large percentage of residuum fuels after taking off the more valuable products, *i.e.*, petrol, kerosene, and lubricating oils. There is then left a certain percentage of light and heavy residuums, of which the latter exists in very large quantities. The commercial oil engine of to-day, with a few exceptions, is not capable of dealing with this heavy residuum, and I therefore claim that when the commercial oil engine is in the position to burn the same grade as is burned to-day in oil-fired boilers it will certainly be a case of wasting fuel if oil engines are not installed, but not until.

The Author: I did not refer to one type of governor more than another. I said there were hit and miss governors, also centrifugal governors, which controlled the speed of the engine by varying the charge. In my experience I have come mostly into contact with the hit and miss governor on the solid injection engine. As to the centrifugal type, I have had to do with a number of boats so fitted, but I cannot say they were any better than the others, judging them by their running. In regard to starting by means of exhaust gases it is quite a common method with two-stroke engines to use the starting valve, after the engine is started up, as a non-return valve between the cylinder and the receiver. After the explosion has taken place in the cylinder you get a certain amount of burnt gas by-passed into the receiver. It is probable when starting up a single or two cylinder engine when combustion chamber is not quite hot enough and a mis-fire occurs to get a certain amount of gas by-passed into the receiver. At the same time I must say I have never known of any trouble arising through that, nor have I ever known of an explosion occurring in the receiver through this cause. The principal trouble was due to the valve itself sticking. Speaking of the engine starting off the small air bottle, that is on an installation fitted with reverse gear, and a hand air pump only is supplied to obtain the initial start. The man in charge makes sure all is in order before starting up. He sees that the boat is properly moored up and puts the engine on load for a time, and while he is having his dock trial he has his receivers charged up. When he is satisfied all is in order he proceeds on a trial run. It should not be necessary to stop the engine. all stopping of the boat, etc., being done with reverse gear while the engine runs free. I do not think there is any other point I have omitted, except in regard to sketches. I did not bring any down with me for this particular lecture as sketches required for the installation of an engine on one particular boat might not be suitable for another. At the same time irespective of the type of engine it is most essential in connection with the installation of an engine that the seating should be securely fitted and secured on to the framing of the boat.

A Member: In the early part of the paper, under the heading "Installation," the author mentions that the blue print shows the "line of shafting, position of motor and reverse gear, also position of fuel and lubricating oil tanks, exhaust piping, sea-cock, circulating inlet and discharge pipes, the latter in many cases being led into the exhaust pipes." Is that the exhaust gas or is that the water circulation that he is speaking of? As to line of shafting, I did not know that the well in many of the boats was timbered. But what is the width or length of the well that the propeller works in when you need to push her stern shaft in from the stern post side? They must be very small boats when they have a movable coupling keyed on. I think in sea-going practice you would have a solid coupling, as you could hardly fit the one described on the tail shaft. Whatever the line of shafting may be, some day the alignment is bound to get out of the true, and therefore the strain would not be just on the bolts, but on the key, and instead of having to renew a coupling you would have to renew the shafting and coupling besides, because the wear and tear would be on that fitting and certainly the coupling would cause anxiety if slackness resulted some distance from port. Why should there be such difficulty with the seating for a motor engine? The motor is not made of aluminium, but of cast iron. If there is a good solid bedding surely there cannot be that great strain, whether it is a timber or an iron seating. There are many timber boats which have been fitted. There were one or two in my own experience where we put a Daimler petrol engine into a boat which previously had had a steam set, and we had no difficulty. The flooring can be stiffened up to the required degree, and I do not think it is such a difficulty as emphasised. In regard to sea-cocks and the large size—I do not see any mention here with regard to grating or strainer being fitted in the mud box, or only on the seacock at the water side. The term "manifold jacket" is new to me. Is it a double jacket or an overlapping jacket? Referring to the oil and the water pumps, certainly a single-acting pump is very simple but has a way now and again of giving out altogether. What is the objection to the double-acting pump? It seems to me that a small centrifugal pump is quite suitable for the job, for the circulating water. Why not recommend a storage tank? It will not want to be on such a large scale, and you could have your cylinder cooled by gravity. I am pleased to see something about the two-stroke engines, because I feel that that is the engine of the future, and it is going to be the most flexible engine as an oil engine, and I hope someone will be able to compound it.

A MEMBER: Reference is made to oil and inert gases, but no mention of the oxygen necessary for combustion. I should like to know when taking up air from exhaust pipe where any oxygen comes from. Also what advantage to starting an engine by this hot pipe or hot gases can we get when it is first necessary to run the engine before it can heat this pipe. In regard to carburetting: I take it that is a petrol or kerosene engine. In regard to heating the air by taking air from around the exhaust pipe, that heats the air to go through the carburetter. Is not the fact of heating that air with a view to giving better carburetting likely to affect the volume of oxygen going into the cylinder, If it heats the air it expands it and therefore there is less oxygen going in, and the remainder of the gases are inert gases. Does not this suggest room for consideration of research regarding too much air being used in ordinary circumstances. The author has mentioned various methods of lubricating the crank chambers, but he does not mention anything of automatic lubrication, whereby the oil falls from a reservoir through a pipe leading to oil sump in crankcase. The reservoir, of course, is sealed, but as the oil is used away from the sump it allows an air bubble to go up through the oil pipe end becoming exposed and that displaces a certain amount of oil in the tank. This, of course, is only an adaption of the old-fashioned water cup fitted to bird cages. The author does not mention what sort of lubricant he uses, and that is one of the great faults of these small engines. It should be quite a simple thing to remember that oil for internal combustion engines should be that which will remain a lubricant at the highest temperature possible, therefore there is a difference between this cylinder oil and cylinder oils as used in steam engines. I should like to ask what is the reason it is necessary to have two sparking plugs with dual ignition when it is quite possible to secure a plug from the cells for magneto on the one plug. What effect do inert gases have on starting? As to economy in oil, we have been talking about economy in the use of oil for a long time. The other day I picked up a book entitled "Coal Oil and Petroleum." This was published in 1865, and they at that time were advocating economy in the use of oil owing to apparent scarcity; yet under their feet were the great oilfields of the future years. We have had experience of people preaching this ever since; surely they do not know the conditions of the oil trade, because if we do not use some of these heavy oils for furnace work we should have to throw them away. Regarding electric starting, I think there is quite sufficient to put everybody off the use of some of these small engines if they have all these directions to read as to starting some of these engines. What can be more simple than an electric motor-dynamo with a chain running over it. All you would have to do would be to put a foot on to the switch and it starts it as a motor. When your cells are in a certain condition it automatically cuts it out as a dynamo.

The Author: My experience has been that when I went to fit out an installation I was always provided with blue prints, and I have noticed that others similarly engaged had likewise been provided. These blue prints gave a general idea as to position of the machinery, but they were not binding to half an inch here or there. Another point brought up was in regard to seacocks and discharge connections. Perhaps my paper is not quite clear on that point. Referring back to the first page you will observe I say "Also position of fuel and lubricating oil tanks, exhaust piping, seacock circulating inlet and discharge pipe, the latter in many cases being led into the exhaust pipe as near the exhaust manifold on the motor as convenient, when exhaust pipe is led out through the stern or through boat's side," the alternative method being to discharge circulating water direct overboard, should have been added. In many cases in motor boats, even of fairly large power, the circulating water is taken from the exhaust manifold jacket and let into the exhaust pipe to keep the exhaust pipe cool. When that is done there is no occasion to fit a connection on the ship's side for the circulating discharge. Where exhaust gases are led direct up the funnel the circulating discharge is, in that case, led overboard through a connection fitted on the ship's side. The member has also made reference to the method of placing propeller shafting in position, and also withdrawing same. In my experience I have had to deal with a fairly large number of boats fitted with tail-end shafts that drew out through the stern tube owing to the engine and reverse gear being placed as far aft as possible, thereby increasing the cargo space. The method of fixing the coupling on the propeller shaft was similar to the Admiralty method of fitting couplings on forward end of propeller shafts for destroyers. There you have a coupling fitted on to the shaft in such a manner that it is impossible for the shaft to be drawn out when running astern. The method adopted is to bore out the coupling parallel a good fit for the shaft; near the forward end of shaft a groove is cut, into which a ring is fitted (in halves), the external diameter of the ring being larger than the bore of the coupling. At the forward end of coupling the diameter of the bore is increased to accommodate this ring. A key is also fitted into a sunk keyseat in the shaft, sufficient metal being cut out of ring to allow key to pass through. To place in position, the coupling is pushed on to the shaft past the groove, the rings are inserted and the coupling driven back over the ring until the shoulder in the bore comes hard up against the ring, when the key is driven home and cut off flush with face of coupling. As to motor seating, when all is said and done there is no difficulty in fitting a seating to hold any motor if it is properly done. During the past four years my experiences have been many and varied, owing to the seating being improperly fitted. I remember several instances where boats fitted with motors up to 60 B.H.P. had simply two pieces of wood placed athwartships for engine seating instead of being fitted in a fore and aft direction. Before these boats had been in commission a fortnight, several of the bearings required renewal owing to the vibration. As the time could not be spared then to fit proper seating, the bearings were overhauled and the boats put back into commission, but before two months had elapsed the shaft carrying the cone in the gear box had broken owing to the excessive vibration. The only remedy was to lift out the motor and fit proper fore and aft bearers on to the framing of the boat and securely bolting them to place. When that was done we had no further trouble in that respect. *Re* strainers for seacocks: if I remember rightly, the member spoke of an elbow on the outside. On the boats I have in mind we had simply a grid plate with a lip cast on the aft side of the plate, projecting out half an inch or so. These grids were fitted over the suctions of all the faster boats, some of which were 20 knot boats, and gave very good results. These fast boats were fitted with gear pumps, and a pump used to last from three to four weeks, when it had to be replaced with a new pump.

Re strainer connected to seacocks. It is very necessary, especially with a large number of modern motors, when there are no great facilities for cleaning out the jacket, to have a mud box or strainer between the seacock and the circulating pump, otherwise cylinder jackets will get filled up with silt and overheating and cracked jackets will very likely follow. A member also asked about exhaust manifold, as the term was new. Exhaust manifold is the term usually given to the connection which joints on to the cylinders, and into which the exhaust gases from the different cylinders are led away to the exhaust pipe. On petrol engines it is water-cooled. Paraffin engines are usually fitted with a non-water-cooled exhaust manifold, as the vaporiser is usually embodied therein.

Re double acting pumps for circulation purposes, when a double acting pump is fitted, one side of the pump only is used for circulation purposes, the other side being used to pump out the bilge, thereby saving a separate single acting pump for that purpose. I have found when a double acting pump was fitted to serve both purposes (circulation and bilge), that a large amount of the oil, which is always present in the bilges, gets past the pump plunger and eventually finds its way into the cylinder jackets, often with disastrous results.

With regard to tanks for the circulating water. I have never seen an installation fitted with tanks for cooling purposes. The water would require to be pumped into the tank in the first instance, and then circulated through the jackets, therefore the existing method is the best. He also spoke of the merits of the two and four-stroke engines. I think I am correct in saying (talking of semi-Diesels) that the two-stroke predominates, but leaving this paper and going to higher powers, such as the large Diesel engined ships, there are not a great number of two-stroke large Diesel engines running. However, I believe an American firm have under construction a large two-stroke engine, and are very optimistic about the results they hope to attain. I understand it will be the largest two-stroke engine that has been constructed up to the present.

Re the remarks on starting up with compressed air, is this meant to apply to the exhaust gases for starting up?

MEMBER: I was rather struck by the lack of oxygen so necessary for combustion. I did not refer to any particular engine, but just to the method of mixing the air for combustion.

The AUTHOR: I have referred to the air supply in a former note showing how it is made up with fresh air. Lubrication gravity feed for a four-stroke is quite all right, but for the two-stroke engine something more definite than gravity feed must be used.

The MEMBER: I mentioned splash feed.

The AUTHOR: You cannot use splash feed in the two-stroke engine. The bearings must be fed independently. By using splash lubrication in the two-stroke engine, the oil would be carried over with the scavenging air into the cylinder.

The MEMBER: But you are talking of four-stroke engines in this paragraph.

The AUTHOR: I have already said the gravity feed is quite all right for the bearings of four-stroke engines. By using splash lubrication for bottom ends, etc., that method is quite good provided the engine is installed with little or no rake. Should the rake be excessive the oil collects in the aft end of crank case, and you have no end of trouble with your forward bottom ends and main bearings; therefore I found that forced lubrication was more satisfactory when you have an engine installed in this manner.

I have not touched on the quality of the lubricant, but I agree it is cheaper in the end to use a good quality of lubricating oil rather than to work with a cheaper grade and run the risk of bearing troubles.

Allusion was also made to two sparking plugs being fitted per cylinder. There are several types of magnetos that can be fitted in conjunction with batteries that only require one sparking plug per cylinder. The reason I mentioned two plugs was the type of motor so fitted that came under my observation, which had the ordinary magneto battery and induction coil.

*To begin with, and taking a single cylinder engine for example, you crank the piston over the top centre, and assuming you have the receiver charged up with inert gas, you admit this gas to the cylinder by means of a hand operated valve, which gives an impulse to the piston and drives it on its downward stroke, the momentum of the flywheel carrying the piston over the compression stroke. When the piston is near the bottom on the downward stroke it uncovers the exhaust ports and the inert gas escapes to the atmosphere. The scavenging air from the crank case sweeps into the cylinder and is compressed on the upward stroke of the piston. When the oil is injected near the top of the stroke you have the necessary mixture for ignition. With regard to the quality and economy in the use of lubricating oil, if you have forced lubrication in a motor of the four-stroke type, that oil is used over and over again, and you have only to make up the waste in the crankcase from time to time. With the two-stroke I have never known during my experience of oil being used over again, because generally it is exceedingly dirty when recovered from the crankcase.

I have never seen an electric starter of the type mentioned, but I've had experience of quite a number of others. These were all of American origin, and provided they were handled with proper care, would last a considerable time. The greatest trouble we had with them was the teeth of the pinion on the armature shaft getting chewed up owing to the armature revolving rapidly before the gear was into mesh. This was due to the drivers pressing down the pedal switch too rapidly, which

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* This refers to a two stroke semi-Diesel engine.—Note the combustion chamber must be heated previously.

caused the armature to revolve, before it had been drawn magnetically into position. Given due care, the electric starter worked very well indeed. A small generator was fitted to the engine for charging up the battery, which was required to drive the self-starter.

NOTES.

The Internal Combustion Engine. Practical Notes on the Installation and Running.

A Member writes as follows, on this subject:—

After reading the report of Lieut.-Col. Lamb's paper on "The Installation of Petrol Engines, etc." in the June Transactions, it occurred to me that the experience with motor launches at Mudros Repair Base might be of interest, as it bears out the Author's remarks regarding the trouble that can be caused by the engines in wooden hulls of motor boats not being correctly aligned when first installed, and the advisability of checking the alignment from time to time if there is any chance of the hull altering its shape. The launches referred to were 100ft. long, constructed of wood and fitted with twin sets of six cylinder petrol engines of 220 h.p. each. Considerable trouble was caused in 1918 through the propeller shafts in some of these boats breaking. These shafts were 29 ft. long of 2½ in. steel, having brass journals for "A" bracket, strut and stern gland—galvanic action being prevented by a rubber ring between the propeller and "A" bracket journal, and copper sleeves between the other journals, the sleeves being let in under the ends of the journals and soldered to them. The shafts were bolted direct to the crank shaft which had a roller bearing thrust between the coupling and end main bearing. There were no gears, as the engines were directly reversible.

When the first fractures occurred, time did not permit of any careful alignment of the shafts, and spare shafts were fitted and the boats sent away at once. At the first opportunity, however, a boat with a defective shaft was taken in hand. Battens were first set up on deck while the boat was afloat (as the decks were strained all shapes) the boat was then placed on the slip and set up until the floating alignment of deck was obtained.

In the first boat dealt with, it was necessary to raise the engines 4½ inches to get a true alignment with the stern gland and "A" bracket. All the remaining boats in the same class were dealt with in the same way as they came in for refit and the engines raised from 2 ins. to 4 ins. Raising the engines necessitated a lot of work in altering water-cooling connections, exhaust outlets, air and fuel pipes, but it meant more trouble to move the stern gland, and lack of tip clearance prevented the "A" bracket being shortened, so that was the only way.

The supply of spare shafts was exhausted and some boats had to be fitted with shafts made of two and three odd pieces joined by muff couplings, but in no case did a shaft break on a boat after the engines had been raised.

The following two mishaps with "Hot bulb" engines may be of interest:—

1. Two cylinder engine. Crank case fractured, frame cracked, and one pair of bottom end bolts broken.

The engine was being started up at the time and appears to have raced. As there was some indication of an explosion in the crankcase, it could not be decided whether the racing caused the bottom end bolts to give way and the damage caused to the frame in this way, or whether oil exploded in the crankcase.

2. Single cylinder engine (pumping). Engine raced in starting up through governor jambing. The vibration broke the holding down bolts and the engine moved, breaking away the extended shaft bearing on the belt pulley, when the engine fell off its seating.

Beyond these two cases the "hot bulb" engine gave no trouble as long as there were good drivers in charge; men who were careful and clean, and who did not experiment too much with their engines.

The "Banjo" ring method of oiling bottom ends is simple and generally efficient but very hard to clean if it does get dirty, and a dirty condition is only detected in most cases after a bearing has overheated.

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SESSION 1921-22.

President: SIR JOSEPH P. MACLAY, BART.

Installation of Marine Oil Engines.

BY MR. WALTER POLLOCK (Member).

Read Tuesday, February 8, 1921.

THE installation of marine oil engines does not usually receive the attention that its importance deserves, no doubt because the conditions are entirely different from those in ordinary marine steam engine practice. As an illustration, many of the most successful oil engines on the market have caused disappointment and have given trouble from time to time because of weak and unsatisfactory engine bearers, while the arrangement of oil fuel tanks is often a source of trouble. When owners and superintendents hear of these and other troubles, they are naturally prejudiced against the adoption of internal combustion engines as the main propulsive power of their vessels, although the oil engines themselves are in no way responsible.

The advantages of oil engines over steam machinery will be dealt with in another paper; it is, however, a mistake to think that there are no disadvantages, and that any engine room designed for a steam plant can be used for oil engines.

The first consideration, in the case of cargo vessels, is the position of the oil engines; although it is usual to fit them as far aft as the space required for the removal of the tail shaft will permit, designers should bear in mind that for any given power, oil engine machinery is much lighter than steam machinery so that the trim of the vessel is affected much more, according to whether the ship is light or loaded. The reduction in the weight of the engines, etc., will in fact often necessitate additional water ballast tanks to give the vessel the same light immersion as a steamer and the same seagoing qualities.

In arranging the machinery space, it should be borne in mind that large Diesel engines call for a considerable number of auxiliaries, and it is necessary to provide ample space so that they can be disposed in accessible positions in the engine room. As most of these auxiliaries are driven by oil engines, they must be placed in such a position as to have sufficient head room to permit the pistons and the connecting rods being drawn.

The above remarks also apply to main engines as regards accessibility, and it is very important that there should be plenty of head room above the engines and strong lifting gear, so that the covers, pistons, piston rods and connecting rods can be taken out for examination quickly, as it will be found that these parts of the engine are usually inspected much more frequently than in the case of steam machinery.

The line of the tail shaft is an important question, and this can be kept lower than with a steam plant of the same power, as the propellers are generally smaller in diameter, except in the case of some of the highest powered Diesel engined ships. The smaller diameter is due to the fact that at the present time the revolutions of oil engines are higher than those of steam machinery. In the case of engines, say 250 h.p., the revolutions of oil engines are double those of the steam engine, but for modern large Diesel engines, they are approximately the same as, or only slightly higher than, those of steam engines of similar power.

Again for any given hull and given power of machinery, the oil engines will weigh less than steam machinery and consequently the displacement when light will be less than that of the hull with steam machinery and the draft aft will also be less, so that the propeller shaft should be kept well down to ensure immersion of propeller.

The lower line of shafting, and the advantage of keeping the propeller as low down as the stern frame will allow, will have the tendency of pushing the engines further forward, owing to the fineness of the lines in the after run of the ship. To avoid setting the engines too far forward, in some cases the shaft line is sloped downwards from the forward end, which inclines the engine aft even when loaded, and much more so when light. This should be avoided as much as possible, because of the trouble with the circulation water, and the natural wear and tear of the after-end of all working parts if the angle is excessive. The best practice, where possible, is to keep the line of shafting parallel with the bottom line of the ship.

When the machinery is fitted amidships, these difficulties do not usually occur, but in the case of twin screw ships, the longitudinal position of the shaft line calls for consideration.

To enable twin-screw machinery to be handled with the least number of men, and to obtain a compact engine room, the engines should be placed as close together as the space necessary for handling them will allow. On the other hand, the propellers should work in water as solid as possible, and to ensure this they must be placed well away from the deadwood. This means that the shaft lines will not be parallel to the centre line of the ship, but this is of no moment, and on many occasions great advantage will be obtained by spreading the shaft lines further apart aft than they are amidships.

For tropical vessels special consideration should be given to the arrangement of the engine room, especially where surface ignition oil engines are fitted, so that the heat from the bulbs will have free escape aft and not interfere with the engineers or drivers or even with the men on deck. Usually the engine room is open above the main deck and a light upper or sun deck is fitted over the machinery space. In the case of passenger vessels, the space between the two decks or between the coaming and the deck above, is filled in with netting.

The design and construction of the engine bearers is of the utmost importance, because oil engines are almost universally single-acting, and the fact that their starting is not so smooth as that of a steam engine must be recognised. It is not necessary to have an expensive construction, but it is essential that the bearers should have greater strength than would be required for steam machinery of similar power.

A short time ago a certain vessel was bought and fitted with two sets of engines of a well-known make, and after the first voyage a large number of renewals were required. It was found that the bearers were designed and constructed of many small pieces of plate and angle so arranged that the natural working of the engine caused " concertina action " and resulted in the engines moving about in a most alarming manner. The engines consequently had to be lifted out and a large sum had to be spent on the bearers to make them good.

The majority of Diesel engines have rigid bedplates, and the engines themselves form a more or less rigid structure.

Many of the surface ignition engines have very shallow bedplates, due to the requirements of the closed-in crank cases, and the necessity of getting to the bottom ends without dismantling the engines. These difficulties in the design, frequently cause the cast iron bedplates to be more or less flexible, which adds enormously to the cost of installation and lining up, and is the cause of much trouble to the bearings should the bearers show the slightest sign of weakness. It will therefore be seen that great attention must be paid to this point.

In the case of engines in steel ships, and having strong bearers, the packing strips should be of hard wood with cast iron chocks fitted in way of the holding down bolts, as is usual in the best steam engine practice.

Bearers and Floors.—The author advocates the floors of steel vessels being carried up from the bottom of the ship to the top bearer plate and right across the ship all in one piece, the longitudinals being intercostal with the exception of the thick top plates and the longitudinal angle connecting the top plates to the intercostals. Fig. 1 shows the suggested design, and the recommended scantlings are given in Table 1.

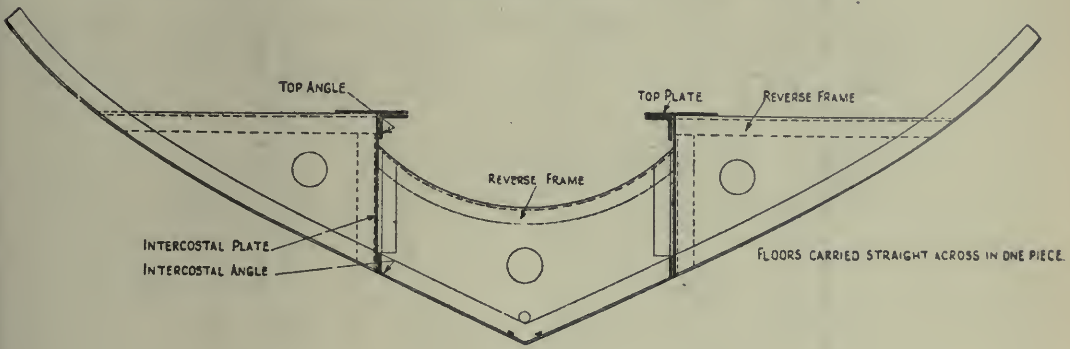


Fig. 1.

The riveting requires careful attention, and the finest workmanship is necessary.

In existing vessels, the floors, reverse frames and keelsons should be removed in way of the engine space, particularly in way of the engine bearers, and entirely new bearers fitted as Fig. 1. There have been many failures and disappointments in fitting oil engines on existing vessels by trying to build up on the floors and reverse frames, instead of carrying out the

TABLE I.

B.H.P.	25	50	100	200	300	500	750	1,000	1,500	2,000
No. of Cyls.	2	2	2	4	4	4	4	4	6	6
Frames ...	1½ x 1½ x ·18	2½ x 2½ x ·25	3½ x 2½ x ·35	4 x 2½ x ·35	5 x 2½ x ·35	6 x 3 x ·40	6 x 3 x ·40	7 x 3 x ·44 bulb angle	7½ x 3 x ·46 bulb angle	8 x 3½ x ·48 bulb angle
Spacing ...	16	18	20	21	22	23	23	23½	23½	24
Reverse Frames ...	2 x 2 x ·25	2½ x 2½ x ·25	3 x 2½ x ·30	3½ x 3½ x ·40	4 x 4 x ·40	4 x 4 x ·50	5 x 4 x ·50	4 x 3 x ·50 double	4½ x 3 x ·50 double	5 x 3 x ·50 double
Floors—Thickness ..	·25	·25	·28	·34	·40	·45	·46	·47	·48	·50
Top Plates, each	6 x ·375	10 x ·40	12 x ·50	12 x ·75	15 x ·75	15 x ·85	17 x ·875	18 x ·90	19 x ·95	20 x 1·0
Top Angles	3 x 3 x ·35	3½ x 3 x ·35	3½ x 3½ x ·35	4½ x 3½ x ·40	4½ x 4½ x ·40	5 x 4 x ·50	5 x 5 x ·50	5 x 5 x ·50	5 x 5 x ·60	6 x 6 x ·75
Intercostal Plate	·18	·25	·30	·35	·40	·45	·475	·50	·52	·55
Intercostal Angles ...	1½ x 1½ x ·18	2¼ x 2¼ x ·25	2½ x 2½ x ·25	3 x 3 x ·34	3½ x 3½ x ·40	4 x 4 x ·45	4 x 4 x ·45	5 x 4 x ·50	5 x 5 x ·50	5 x 5 x ·50

NOTE.—The above dimensions are in inches.

Engine Bearers for Steel Vessels.

recommendations suggested. Fig. 2 shows the arrangement of engine bearers for vessels built of wood, and Table II. gives suggested scantlings.

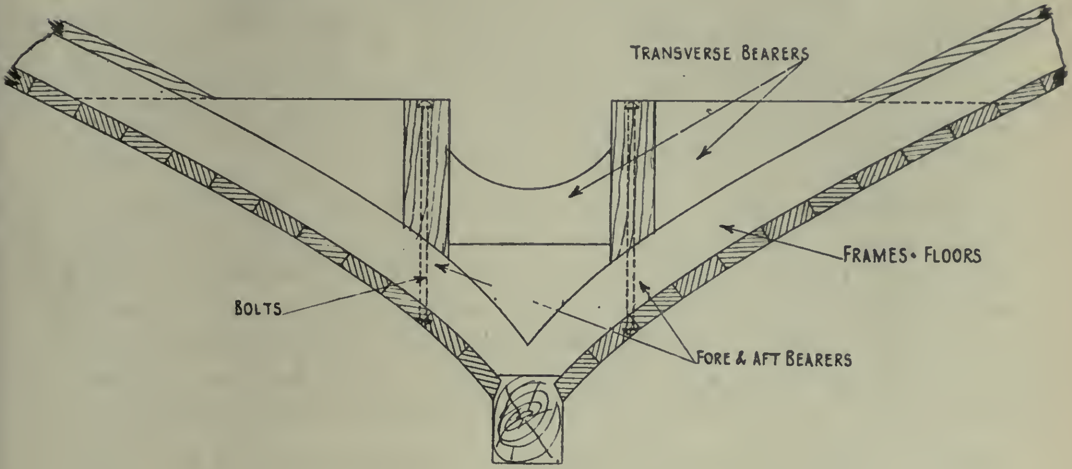


Fig. 2.

TABLE II.

B.H.P.	25	50	100	200	300	500	750	1,000	1,500	2,000
No. of Cyls.	2	2	2	4	4	4	4	4	6	6
Floors ...			6" x 6"	8" x 8"	9" x 9"	11" x 11"	13" x 13"	14" x 14"	15" x 15"	16" x 16"
Frames ...	2½" x 2½"	4" x 4"	5" x 5"	7" x 7"	8" x 8"	9" x 9"	11" x 11"	12" x 12"	14" x 14"	15" x 15"
Spacing ...	14"	16"	18"	20"	23"	26"	30"	32¼"	33½"	34½"
Fore & Aft Bearers Width...	4½"	5"	8"	9½"	10½"	12"	13"	13½"	14"	14½"
Length of Bearers in ft. ...	13	17	25	40	50	60	65	70	75	80
Transverse Bearers Width ...	4"	4½"	6"	7½"	9"	10½"	11"	11½"	12"	12"
Bolts thro' Bearers Dia. ...	⅞"	1"	1¼"	1½"	1¾"	2"	2"	2"	2"	2"

Engine Bearers for Wooden Vessels.

The saving in space which can be effected by the use of oil engines instead of steam machinery of the same power, is so considerable that new conditions and calculations have to be considered for any given type of ship. In the case of Naval vessels, the advantage is very great, because the sub-division of the vessel into small compartments can be obtained with ease and without any loss of efficiency or economy in working.

For vessels that are designed to carry bulk cargoes, additional cubic capacity is easily obtained. In dealing with shallow draft vessels, which are usually of light scantlings, it is important to carry the longitudinal bearers as far fore and aft as possible, and where convenient, for a distance at least twice the length of the engine; in no case should the longitudinals stop abruptly at the end of the engine. A few web frames in way of the transverse bearers or floors, also prove of advantage if they are not in the way of the oil tanks.

Fuel Tanks.—In the case of large vessels, double bottom tanks can be used for fuel, but these should not be carried immediately under main engines if it can be avoided. With smaller vessels, cylindrical tanks are usually found the most satisfactory, as, if well designed, they are not so liable to leak when say half full and being strained in a sea-way. Rectangular tanks should be avoided for anything over 250 gallons.

Besides main fuel tanks which can be usually stowed at the sides of the engine room, it is usual to have a daily service tank, and this should be arranged and marked in such a way that the hourly consumption can be checked when required, and should have connections from all the main fuel tanks. This daily service tank should be fitted in such a position that the bottom of the tank is well above the fuel pumps, and that the fuel pipes may be arranged without any upward or sharp bends, so as to avoid the risk of an air lock.

All main fuel tanks should have filling pipes at least 3 ins. diameter, where possible, connected to the main deck. It is wise to have a large gauze funnel with a screwed end to connect it to the deck filling pipe, so that the fuel tanks can be filled rapidly from the shore pipe connection.

All fuel tanks should have overflow pipes to the deck with a goose neck end and arranged so that they are always open and thus relieve the pressure on the tanks should any expansion of the oil take place. Again, the tanks should be designed with large manhole doors in easily accessible positions to enable the tanks to be thoroughly cleaned out at least once every six months. Connections to the tanks should have thickening or boss plates, so as to ensure a good oil-tight connection. The drain cock at the bottom of each tank should, if possible, be fitted with a " sump," and be placed in an accessible position.

All tanks should be of heavy scantlings with plenty of subdivisions to prevent rapid movement of the oil in bad weather, when only partly full. Electric welding makes the best job, especially for the rings for manholes and the seats for fittings, sump attachment, etc.

The service tanks should be so fitted that in the event of the oil pipes breaking away there is no chance of the oil pouring on to the engine or any of the auxiliaries, and causing risk of fire.

In cold climates, arrangements should be made for warming the oil in the fuel tanks. This can generally be arranged by attaching a heater to the daily service tank, and when the engine is under way the engine room will soon become sufficiently warm to reduce the viscosity of the fuel in the main tanks, and allow the daily service tank to be pumped up with ease. In any case, the pump from the main fuel tanks to the daily service tank should be of simple and efficient design, to give a good pressure on the discharge side, and it should be fitted as low down as possible, so that the oil will flow into it.

In small vessels a semi-rotary hand pump is usually sufficient, though in large vessels a power driven pump is desirable.

Pipe Lines should be so arranged that all joints are in an accessible position where any leakage can easily be seen. Care should also be taken that the valves are fitted where they are easily accessible.

All small fuel pipes should be of copper with ground coned joints. Copper fuel pipes should be re-annealed periodically—the greater the vibration to which they are subjected, the shorter should the interval be.

Drip Trays should in every case be fitted under the main engines and all fuel tanks, and these should have a " sump " with a large drain cock on the lowest portion of it. These "sumps " and drain cocks like those in the fuel tanks should be fitted at the after end, as oil-engined vessels are usually down by the stern, especially when light. In any case, the drain cocks should be so arranged as to allow a can or bucket to be placed under them; if this is impossible, a drain pipe should be connected up to serve the purpose.

Ventilation of Engine Room.—This item requires more care and consideration than in the case of steam machinery; even in the best regulated engine rooms with oil engine machinery a certain amount of oil will find its way into the bilges. It is therefore necessary to have a down-cast ventilator from the upper deck or casing carried right down below the engine room floor to the bilges, and as far forward as possible, the design of floor bearers being so arranged that the air will freely find its way right aft. An up-cast ventilator must also be fitted at the after end of the engine room. In any case, the position of the ventilators should be such that the incoming air will not be directed into the cylinders or engines. Ventilation of fuel tanks is discussed on another page.

Water Drip Tanks.—In vessels fitted with engines which require water cooling of the bulbs to prevent pre-ignition, it is necessary to provide sufficient storage for clean fresh water. These tanks should be separate from the drinking water tank, and placed in the engine room, as high above the cylinders as possible to ensure sufficient head of water.

On no account must salt water be used in these tanks or for "water drip." The consumption of water is generally about the same as that of fuel when running at normal speed.

Deck Machinery.—The deck machinery, such as the winches, windlass, capstan, etc., will, if driven by oil engines, call for the provision of a good supply of cooling water, which, if drawn direct from the sea, may lead to trouble in the case of large vessels, owing to the height of the pump above the sea. In the case of coasters which are frequently loaded and discharged when ashore in tidal harbours, this method becomes impossible, and the best arrangement is to fit cooling water tanks in a convenient place on deck, with the necessary water connections. If these tanks are fitted in an exposed place, the usual atmospheric conditions assist the cooling process; on the other hand the water may get frozen in winter. In every case, they should be so disposed that manhole doors for cleaning and cocks to drain the water off, can be fitted in an accessible position.

The pipes connecting the tanks to the deck auxiliaries should, where possible, be of galvanised iron and should be clipped up to the bulwarks or hatch coamings as far as possible off the deck, great care being taken that the drain cocks can drain off the water in frosty weather.

Deck Control is sometimes required in small vessels. Fig. 3 illustrates a useful method that can be applied to almost any type of oil engine.

Steering Gears that reqiure to be driven by machinery are troublesome things, especially for oil-engined vessels; so far, a directly connected oil engine has not been adopted. The electric hydraulic system fitted at the rudder head is very efficient although somewhat expensive for small vessels.

Some designers have fitted a donkey boiler arranged to burn oil fuel, so that steam steering gear and steam winches, windlass and capstan can be adopted, although the inefficiency of such a method is not fully recognised; however, should it be decided to have a donkey boiler, under no circumstances should it be in the main engine room, but it is absolutely necessary to fit it in an entirely separate compartment, without any direct connection to the engine room in any way; so as to avoid the smoke and soot which are inseparable from an oil fired furnace when steam is being raised or shut down.

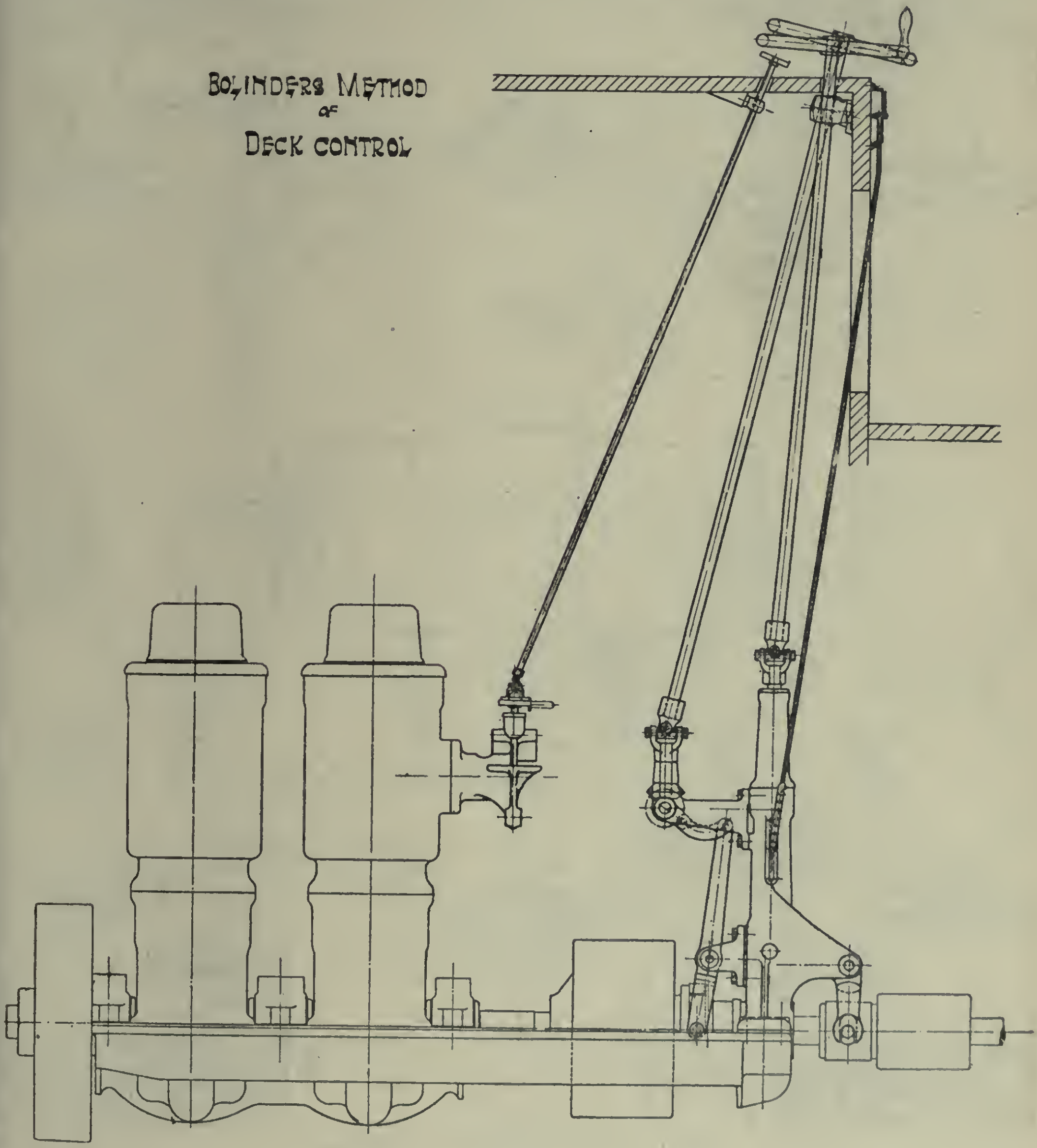


Fig. 3.

Winches.—In this paper, the question of oil driven winches in small vessels up to only 200 or 300 tons will be dealt with. One or two winches, each with a separate oil engine will suffice, but in larger vessels with two hatches on one deck the most economical method is to instal an oil engine in a central house to drive at least four winches, see Fig. 4.

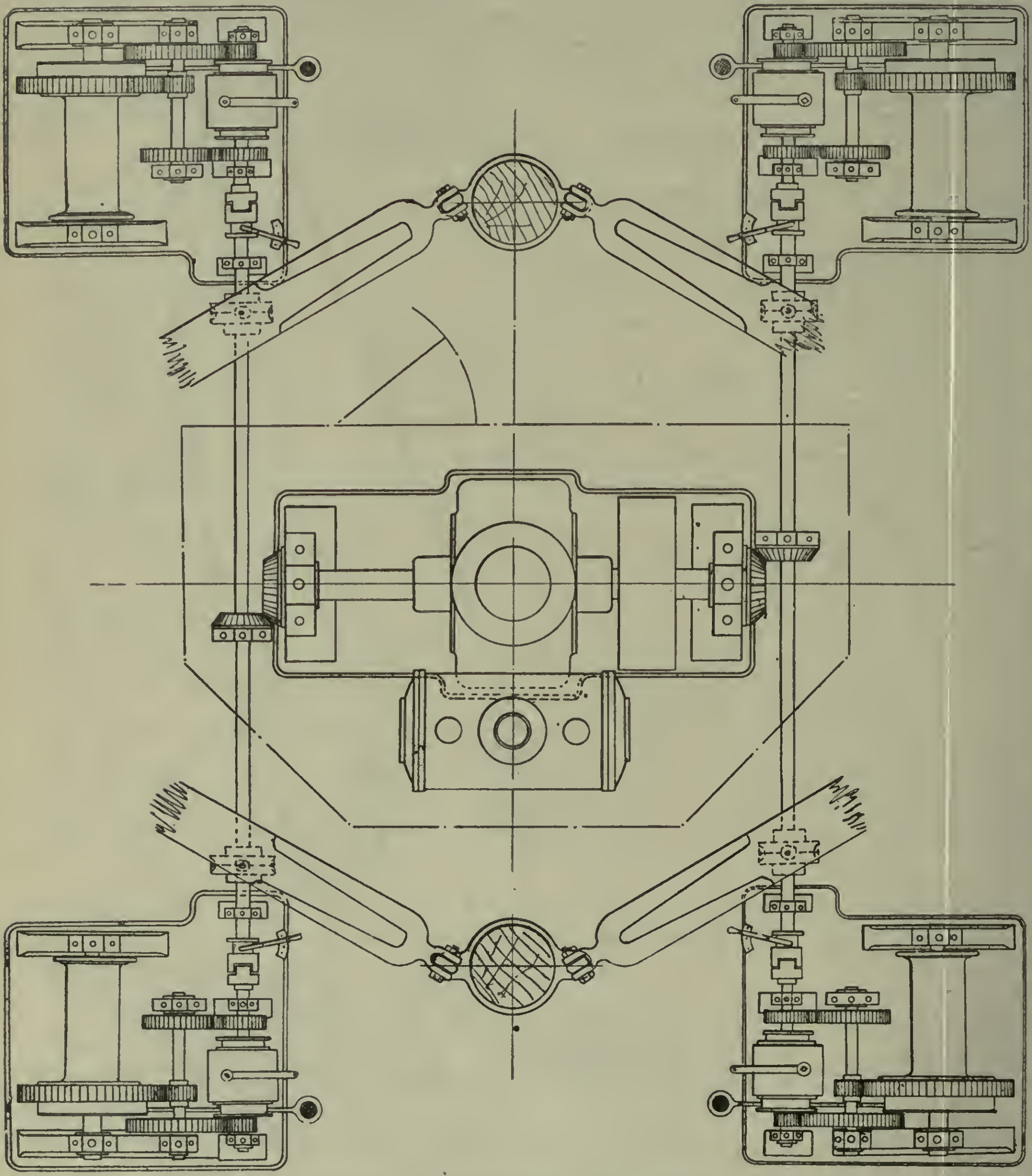


Fig. 4. Economical Oil-driven Winches.

Windlass.—In small vessels of a few hundred tons, it will be most economical to drive the windlass by a messenger chain from the forward winch. In larger vessels, a windlass directly driven by an oil engine is more satisfactory. In this case, the oil engine, and if possible the windlass, should be fitted below the forecastle deck. In any case, the oil engine should not be fitted above the forecastle deck owing to the difficulty of starting in rough and dirty weather.

Capstans can be fitted aft and driven by a separate engine, as the after deck house provides accommodation for the engine or will at least form some shelter.

Air Bottles or reservoirs are necessary in every oil-engined ship, and it is extraordinary to find how many of them are fitted in totally inaccessible positions, although it does not matter whether they are fitted vertically or horizontally.

It is absolutely imperative that they should be of ample capacity and fitted so that all the valves and pressure gauges are 4 or 5 ft. from the platform from which they are operated. Fig. No. 5 indicates a satisfactory method of fitting these air bottles.

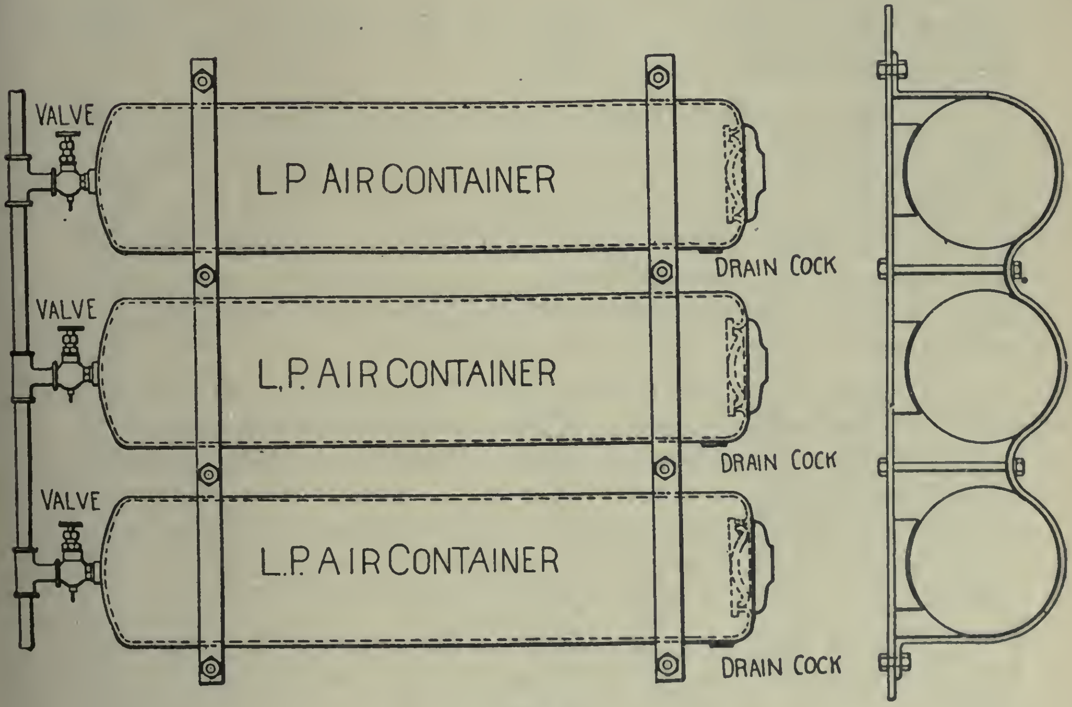


Fig. 5.

It will be noticed that any one bottle can be shut off and removed for cleaning or repair, without disturbing the remainder. For this purpose, it may be necessary for each bottle to have separate clips. If the bottles are of very large diameter, an eye bolt should be fitted in a deck beam above, so that a small tackle can be used to handle them.

It is assumed that all air bottles over 7 ins. or 8 ins. in diameter will have removable tops or bottoms, or alternatively have cleaning doors, as it is almost impossible to inspect and to clean out a bottle built with only a pipe connection at the top and a drain cock at the bottom.

All air bottles must have a drain cock at their lowest part, and this should be used frequently to blow down any water and oil which may have accumulated, more particularly on those engines which charge the bottles with the products of combustion.

In small vessels without air compressors, it is necessary to have an efficient means of pumping up the air bottles for starting; probably the best method is to have a hand plunger pump driven by a crank shaft fitted with a heavy flywheel.

Circulating Water.—The supply for internal combustion engines should be ample, more especially if it is likely that they will be worked in the Tropics.

The water should be taken from straight through cocks well above the bottom line of the vessel, so that the roses do not get clogged when the vessel is aground. In some cases, it is wise to fit weed boxes, or a circulating suction may be fitted on both sides of the vessel with a connecting pipe in between. Shut off cocks should be so arranged that they can be easily opened and shut, and overhauled without disturbing much of the engine room floor.

It is common in most oil engines to pump the circulating water through the cylinder jackets first, then through the cylinder heads, then into the silencer jacket and then overboard. Sometimes the water is also used to cool the air compressor and the main bearings, but this is a complicated and undesirable arrangement, and should be avoided where possible.

In some cases the designer has no option but in others he can arrange that each cylinder and its accompanying parts shall have a separate water circulating system supplied from the one pump. This is of special advantage for vessels working in tropical climates,as the circulation of water from one cylinder to another leads to too great a difference in temperature.

The overflow or discharge pipe on a ship's side should be fitted in such a position that it can easily be observed by the engineer from the deck. In smaller vessels, especially in Scandinavia, it is usual for the circulating water to be discharged into a funnel in the engine room, and thence overboard so as to be in constant view, though of course, this practice cannot be adopted for sea-going vessels.

Silencers.—A cast-iron water jacketed silencer is fitted on the engine, and as has been previously mentioned, there are many types of secondary silencers, the most usual being the vertical one formed of steel plates fitted in the upper portion of the engine room casing, or in the funnel, arranged with a manhole door for cleaning, and a "sump" with a drain cock and pipe to drain to a convenient position.

Some designs of silencers provide for perforated baffles or divisions, others for partial baffles or divisions, but all these should be avoided, and the perfectly plain silencer will be found the most effective and will cause the least resistance to the flow of the exhaust gases.

Exhaust Pipes even in the best regulated oil engines have to convey hot gases, the natural outlet for which is a funnel. All exhaust pipes should be attached to a silencer usually fixed to the engine. Where possible, the exhaust pipe should be carried vertically and into a secondary silencer, thence to a funnel.

Under no circumstances should the exhaust pipe be reduced in area, in fact it will be found good practice to make the internal diameter 10% greater than the outlet on the silencer.

Bends in steel exhaust pipes should have a radius of at least five times the diameter of the pipe. In the case of yachts and passenger vessels, light cast-iron exhaust pipes, water cooled for the whole length, are sometimes fitted.

Several installations, especially in small vessels and auxiliary sailing ships, have had horizontal exhaust pipes carried right out to the stern. This practice should be avoided where possible on account of the risk of sea-water getting into the exhaust pipe, and in due time forming a deposit, and the difficulty of cleaning the pipes, while the heat given off, causes trouble if the pipes pass through the accommodation. Where a funnel is not required for the sake of appearance and to obtain a "steamer's turn" in loading and discharging in port, a simple exhaust pipe can be carried straight up from the first silencer through a secondary silencer, and discharge into the atmosphere above the casing. This is a specially convenient method in the case of twin-screw vessels. In the case of sailing vessels and yachts, a satisfactory method is to carry the exhaust pipe well up one of the masts, but it should be kept a few inches clear and large iron plates should be fitted in way of wooden decks, and bulkheads and air casing plates or outer pipes round the intermediate parts, to prevent risk of fire or radiation of heat.

The exhaust from oil driven deck auxiliaries cannot be dealt with in the same way as that from steam auxiliaries. Generally, it will be found most advantageous to carry the exhaust pipe up the mast or casing 10 or 15 ft. above the deck. To prevent noise, these pipes should be of larger diameter than the exhaust outlet on the engine. They should also be secured firmly to the mast or casing. A drain cock of large diameter should be fitted at the bottom of the pipe or at the lowest point of the silencer, so as to draw off any water which might have accumulated owing to rain or bad weather.

Funnels are usually favoured, especially in coasters and tugs, as the owners think they make the vessels look as if they were fitted with powerful machinery. The funnel can be used to accommodate the secondary silencer and also to act as a ventilator to the engine room or other parts of the ship.

Bilge Pumps are essential for all vessels that go to sea. The installation should provide for one driven from the main engine and also one as a stand-by, driven from one of the auxiliaries, the pipes and connections being arranged in the usual way.

The deck water service can be supplied by the bilge pump if a water connection to the ship's side is provided; the usual care should be adopted in fitting non-return valves in way of the bilge suction so that the bilges cannot be flooded from the seacock.

Overboard discharge pipes, if placed near the water line, generally have non-return valves fitted to prevent flooding in the event of fracture.

Engine Room Fittings.—A whistle is required on deck, as in steamers, and the air for this should be taken from the low pressure containers. If only high pressure containers are fitted, a reducing valve should be used. The whistle should be of extra large size, and if possible should be adjustable, so that it can be used with any variations in the air pressure.

Fuel oil filters should be fitted between the daily service tank and the fuel pump, the fuel pipe being so arranged that air locks are impossible. The most economical type of filter is one of cast iron about 10 ins. in diameter with a loose cover for taking out and cleaning the filtering medium.

In a number of engines, a fuel priming cock is fitted on the side of the filter to pump the fuel oil and air through the pipes.

The engine room floor, benches and racks should all be of steel, wood being entirely dispensed with, except perhaps for the top of the engine room seating.

Sand boxes and fire extinguishers are recommended for all types of oil engines, both being fitted near the engine room entrance, the fire extinguisher being on deck near the top of the engine room ladder.

Guards are necessary and advisable, especially in way of the fly-wheel and pump gear, but they should be so fitted that they are easily removable.

To make an efficient installation, it is very necessary that proper steps and platforms should be fitted to enable the engineers to get at the various parts, the tops of the cylinders, the pressure gauges, the air valves and connections, the cocks to the service tanks, the lubricating oil tanks, etc.

The tail shaft should, like all well designed jobs, work in oil or grease, and the gland at the after end should be so designed that it will not get damaged when a rope or chain gets round the propeller or the end of the tail shaft.

Dock Authorities.—As this paper deals principally with engines that use safe fuel oils, there is not much doubt about the installations of such vessels passing any Harbour Board or River Authority, but it is well to consult the Harbour Engineers of any particular port that the vessels have to work out of.

Classification.—There are not at present many rules concerning the installation of oil-engined vessels issued by the various Classification Societies and Board of Trade, although it is as well to get the latest rules from the Societies in the case of vessels being "classed," and the Board of Trade rules for vessels that are destined to have passenger certificates.

Cleanliness should be considered by the Naval Architect as well as the engineer. All ledges or shelves should be avoided where possible, so as to minimise the number of places where dust can rest. Where these horizontal surfaces are essential, they should be so arranged that any angle stiffeners, both fore and aft and transversely that may be necessary, are fitted on the under side, so as to leave a smooth surface on top for cleaning purposes.

The engine room ladder should be fitted clear of the engine. If it is absolutely necessary that it should cross over any portion of the engine or shafting, the ladder should be plated underneath for its full length, to prevent the smallest amount of dust or dirt getting on to the engine. Proper mats either rubber or coir, should be fitted outside the entrance to each engine room, also inside on the grating platforms and again at the foot of the ladder.

Simple arrangements should be made to enable ventilators and skylights to be covered over or closed quickly when in the neighbourhood of any form of dust, such as when loading or discharging, or when near other vessels that may be coaling or dealing with dusty cargoes or when near a quay where the wind may be blowing the dust off the roads or quay.

Lubricating oil tanks, drums and the filling can should all be arranged so that it is impossible for the slightest amount of dirt or grit to get into the oil, and even if it does, then the method of filtration should be such that it is impossible for it to get to the oil boxes of the engine.

Engine rooms are not cleaned out as frequently as they should be in oil-engined vessels, and it is not uncommon to find the bilges in a very dirty state; in many cases these troubles arise through faulty installation. If engineers and drivers would appreciate the importance of having an absolutely clean engine room under every condition, they would have much less trouble in the long run and the oil engine would soon have a better reputation for good and efficient work. A number of installation plans of small vessels are laid on the table and details will be explained with pleasure.

A Member: The author rightly points out the necessity of taking great care in the bearers fitted in the ship for holding down the engine seating which requires more special attention than an ordinary steam engine installation, and I expect want of this has caused trouble occasionally, especially when fitting oil engines into old vessels. We are not told much about installing oil engines into wooden vessels, probably as these are getting out of date; yet a deal of trouble has arisen when care was not taken to fit proper bearers for the sole plate, and I believe defects arose through fitting thwartship, instead of fore and aft bearers. He has given us many points and a good idea of what should be done to ensure a good and well-working installation; and the latter part of his paper, in which he lays great stress on cleanliness and avoidance of dirt, can be taken to heart and fully regarded by all who are running oil engines, as there seems a deal of carelessness in respect to this, and it is one of the worst enemies the oil engine has, especially in small vessels.

A Member: The completeness of the paper robs us of the chance of much criticism. I should like more detailed information in regard to steering gears. I have to deal with a class of vessel, working on the Thames, in which the steering gear plays a very important part—almost as important as the propelling machinery. Everybody, I think, knows the duties of these smaller river vessels. The duties of these engines are very heavy and exacting, and I should like to know how he proposes to deal with the steering gear, where it is in constant use. Without steam steering gear a lot of the work that is at present carried out would have to be left undone, and for that reason a good steering gear is an absolute necessity for this class of vessel. Reference is made to steering gear, but if the author would give more detailed information it would be of considerable value.

A Member: In regard to the first paragraph in the paper—I think engineers at the present day are quite confident of the oil engine giving good service, but there is a doubt about the reversal of the oil engine on the part of marine steam engineers. Oil engines have been brought to great perfection, but there is a feeling that they are not so flexible as steam engines; with further experience of the oil engine we shall get more confidence in them as we go along.

Then comparing the weights of the oil engine with the steam engine (less boiler plant), what is the average weight per horsepower of the oil engine, I mean the weight of metal in the construction of the engine. In an engine room constructed for an oil engine, I apprehend there would be plenty of head clearance, as there is no necessity for midship bunkers. In a well-designed engine room we should have ample clearance for drawing any piston or overhauling machinery, and that reminds me to ask: Are the pistons drawn with piston rods or are they pistons with trunks? I understand, in gas engine practice, the trunks are drawn from the underside, or crankside of the cylinder. What is about the average piston speed of these engines, up to 250 horsepower? They have about double the revolutions of the steam engine. What is the behaviour of the oil-engine driven ship when "light," and more especially when these engines are aft, as near to the stern as it is possible to get them? I take it they cannot be placed any nearer to the stern-post than a steam engine. I agree as to the scantlings for holding the bed-plates down as shown, but I think a little might be saved in the transverse girdering, angles and plates, to avoid too many enclosed spaces, so that everything might have a wash.

We cannot get away from gases being thrown off from the oil used, and there is always the danger of a naked light, as bilges are not ventilated to the preventative degree. With regard to the oil tanks I always thought the cylindrical tank was the cheapest to make, and the safest. But I do not see any provision for cleaning it out, except a large manhole door. Unless you have a method of washing out before re-filling with oil, risk of dirt becomes greater. I do not know whether this oil leaves any scum about the plate. These are little points well worth taking the advantage of asking questions about, seeing the author of the paper is present to give us an opinion on them. Then there is the deck gear. We have at the present day, and have had for some time, good submersible motors for deck gear. Instead of spreading oil engines all over the ship so that we have, as some people tell us, an engineer's shop in every hole and corner, why cannot we have something of a stable engine? I do not think the cost would be so heavy. Then let this engine drive a generator in the engine room, and motor it in from that to any auxiliary you may have. There would be no question of cooling water, no question of Baltic terrors, of getting frozen up and split pipes.

Steering gear was mentioned. To have that electrically driven and controlled would, I think, be an advantage to the oil-engined ship.

A MEMBER: The author in his paper seems to lay great stress on the necessity of having the donkey boiler in a separate compartment of the ship, away from the engine altogether. The reason he gives is on account perhaps of a little dust getting on to the machinery. But surely, with this class of engines it is not absolutely necessary that no dust at all should settle on them. I have been aboard several ships on which the donkey boilers are in the engine rooms. These were oil-fuel boilers, and no trouble had arisen. It seems to me that if the internal combustion engine has to be guarded against every little bit of dust that may blow into a ventilator when working cargo, it is hardly right, and in my opinion, these engines should be made so that the dust should not interfere with the working of them. I should like to know if the author has any other reason why the donkey boiler should be placed in a separate compartment.

The **Chairman**: The remarks made by the author on the question of silencers interests me, for he tells us that baffles should be avoided, and apparently he would take the baffles out, whether wire gauze or plates perforated with holes. I agree with him, and I recall my experience with a car engine of 50 horsepower. When trouble arose I suspected the silencer, so when the car came in, the silencer was opened out, and all the plates were removed. Later, the owner was take for a ride and he made no suggestion that there was more noise and as the car pulled very well he was pleased. I tried another silencer. Instead of a long tube three or four feet in length, I arranged a series of short tubes, and they acted as a series of short expansion chambers, and it was much quieter. The ideal silencer is an apparatus in which the volume of the exhaust gases is continuously being increased so that the pressure falls continuously down to that of the atmosphere. The trouble of noise arises when the engine exhausts directly into the atmosphere. Then gases issue at high pressure, striking the air suddenly and so gives rise to the noise. If the gases from the exhaust valve pass into a chamber with sufficient volume to act as an elastic buffer, and so take up the kinetic energy of the exhaust gases, the expansion down to the atmospheric pressure can take place without any great noise. There is another point raised about dust. Now the good old steam engine will go right on to the last kick. We can get her going with slack in the bearings and other defects; but she will keep turning round. But the oil engine does not act anything like that. It just stops, and that leads men like the author to take and advise the greatest precautions. In the case of a motor car, you will sit down and examine the carburetter, and having done that you are conscious of having done nothing beyond making your fingers greasy. But she starts up, and runs merrily. You stop again, and look once more for a piece of dust in the carburetter, and the vibration of opening out will usually jar out the elusive speck of dust. I feel that every precaution should be taken for the avoidance of such troubles. At sea you are not troubled with carburetters, but because one trouble is removed if often follows that there are four or five others—if not more—equally troublesome to contend with. Another point was the question of propellers. The author speaks of an inclined shafting, and his advice is not to have it. He says: " The best practice, where possible, is to keep the line of shafting parallel with the bottom line of the ship." I am not possibly quite up to date with regard to the latest experience in the use of inclined shafts, but I remember, in the case of a certain Herreshoff vessel which was brought over here some time since, that the shaft came out below the rudder and went two or three feet astern; so that the propeller was really in solid water. That arrangement could only be used in a vessel which was always going to run in deep water. If she were near the land she would touch the ground with her propeller first, and crumple it up. He advises that, with twin-screws they might open out towards the stern. It seems to me that in doing so we are running the risk of the propellers standing out and becoming a nuisance when lying alongside a quay or wharf. Perhaps he has some qualifications to this, because that is a point that cannot have escaped his attention. That the propellers should work in solid water is an undoubted fact, because nothing would spoil their efficiency so much as to work in troubled water. Otherwise cavitation would not hurt the propeller. The paper contains quite a number of practical features that mark it out as one that is written by a man who has first hand knowledge of his subject.

A MEMBER: The question of compressed air starting has been referred to, and the loss of air, also to the lot of labour in getting back the pressure. I can sympathise with this view from similar experience. I recently saw an old bell-crank type compressor installed in a new vessel for emergency compressor service, when one would look for an improved type of a rotary design of which I believe there are several makes available. Some forms of rotary air compressors I have seen will pump up air pressure quickly and in one instance the fitting of a rotary compressor with connecting hose on a river-side wharf was a great saving of time in helping out cases where engine starting air had been used up. Another point in the paper was in connection with water drip. One had hoped that water drip was disappearing, but it seems to be in as much use as ever with the general run of hot-bulb engines. I agree with the author when he says not to use salt water; and one has to be careful not to use water with solids in it. I have heard of cases where cylinders were badly worn from gritty deposits left in them by the water drip. It may be an advantage to use distilled water or rain water.

MR. W. POLLOCK: A number of the points in the paper may be considered as elementary. It is rather strange, especially in small ships, to find small but important things overlooked. A member raised a question in regard to steering gears. In so doing he has raised a question in regard to one of the most difficult problems we have to deal with. I take it from what he said, that he was referring to tugs. It is a very difficult proposition, that of steering properly, and at the present moment we only have hand gear for oil-engined boats of moderate size. But I do not doubt we shall ultimately use some form of hydraulic or electric steering gear for the vessels he has in view. Up to, say 800 horsepower, I do not think a direct-coupled oil engine is a practical proposition at the present time, because of the work, and intermittent load. That subject, I may mention, is dealt with in the next paper I am reading, on Auxiliary Machinery. Other than the hydraulic or electric steering gear, one might have air, using the air compressor to supply a reservoir. I think it is quite possible, when the engine is not exposed, to obtain a steering gear worked entirely by air. The efficiency, I believe, is very much greater, if not exposed, than is generally believed. A friend of mine has this proposition before him, and is experimenting. If he obtains efficiency he hopes it will also apply to winches, capstans, and other fittings. A member referred to auxiliary compressors. I do not know quite what size he had in his mind. Of course, with very small engines you only have a more or less glorified motor car pump. Then we have various compressors worked by hand; but as I have mentioned in the paper, the best method is to have a hand plunger pump driven by a crank handle fitted with a heavy fly-wheel. Then you come to the next stage, and that is, that you should have a power-driven pump. There are a number of power-driven air compressors, which I shall mention in detail in my next paper. You can deal with it by having it driven off the main engine direct, or, as I have done on many occasions, through a clutch. You can drive them by belt off a small fly-wheel, or you can have an entirely separate engine. One of the difficulties we have to deal with in installations of, say 150 h.p., is the cost of a separate air compressor driven by a separate engine. You can get a fairly satisfactory arrangement by using a petrol or paraffin engine, but it is not desirable. To get a heavy type of engine driving the air compressor becomes too expensive for the ordinary installation. I have found from experience that the main oil engine is already quite expensive enough. A very good method is to have the air compressor driven by a separate engine, and have a coupling on each side, so that an air compressor can be driven from one end and an electric light plant and/or bilge pump from the other end, so that you have a combination auxiliary set. Many alternatives could be adopted with one small engine. Another member raised several interesting points. I did not quite follow all of them, because I was not sure what size of engine or what size of machinery he had in view. With regard to weight per horsepower, that is a very varying quantity. In the case of the Diesel engine it depends whether it is a 2 or a 4-cycle engine; and in my view the weight of the engine is of very small consequence when you have to take into consideration the enormous weight of auxiliaries. Taking the ordinary small type of engine, say of 350 horsepower, of the hot-bulb or surface ignition type, you will find it will weigh about 25 tons, or 14 horsepower per ton. That is the engine complete up to the end of the shaft; and the engine requires little or no auxiliaries. With regard to the removal of trunk pistons, there are very few, except fairly large Diesel engines, that can be removed at the lower end, and if you can remove the trunk pistons in that way there is greater risk in handling and consequent damage than lifting overhead.

I think it is generally found more economical to take them through the top. With regard to piston speed—this, of course, varies with different engines; good practice is for an engine of 350 b.h.p. to have a piston speed of 675 ft. per minute. He also raised a question in regard to deck gear and motors, mentioning the engine room auxiliary in the form of an electric set. These questions are being dealt with in a paper already written to be read next month. I am dealing with the same subject in several aspects, and I shall look forward to having criticism when the paper is read. In regard to having a separate engine compartment for the donkey boiler. In a Diesel engine job it is not so necessary to separate the donkey boiler from the engine-room, but there are very few—I do not know of one—Diesel engine installations from 2,000 to 5,000 horsepower, that has not a small auxiliary engine, usually of the surface ignition type. That small engine is a sufficient reason for doing away with the practice of having the donkey boiler in the engine room. It is a very nice proposal to suggest that the engines should be made so that they will not be troubled by a small amount of dust, but if we carefully examine any of the well-known makes we shall find the degree of accuracy of workmanship is astonishing. It is because of that accuracy that the least possible amount of dirt will cause trouble. It is really a simple matter, to put the donkey boiler in some other position, and the object of it all is to get an installation that will give as little trouble as possible. In a record of 120 cases of trouble in surface ignition engines up to 300 or 400 horsepower, it is no exaggeration to say that 30 per cent. are due to dirt and dust, either in the engine room or in the lubricating oil. In many cases of boats working in river service I think it would be more correct to put the percentage at 60 per cent. The Chairman raised the question of the silencer with short pipes. There are many forms of silencer. One I remember with five or six fairly large tubes, but the difficulty we found was to clean them. It meant taking off the whole of the end of the silencer and the exhaust pipes and individual cleaning. If you have a silencer of the ordinary open drum type you can, with the smallest amount of heat in the engine room, bring the oil deposits down to the bottom of the silencer, and if it is arranged with a sump, as suggested in the paper, it is not very difficult to draw it off. It is quite a simple matter to take off the manhole door and put your arm in and clean the deposit of soot and oil out. One great difficulty is not only the back pressure of the exhaust, but a far greater difficulty is to keep the silencer clean. In one case I know of a silencer which had a series of holes $\frac{1}{2}$ in. to $\frac{5}{8}$ in. diameter. Measuring these holes up we found we had a total area of at least 60 or 70 per cent. in excess of the area of the exhaust pipe. But that did not prevent the top of the silencer being blown out, as the accumulation of deposit and oil due to bad combustion, etc., was so great that there was practically no clear area through these holes. We had all these division plates removed and the silencer put back, and although there was slightly more noise it was assumed the engine was working better and there was no further trouble with the silencer. With regard to the Herreshoff inclined shaft—I remember the case that the Chairman has mentioned, and do not think the efficiency had much to do with the inclination of the shaft. It was due to the more thorough immersion of the propeller. That, of course, is a very nice proposition, and is quite satisfactory in yachts and demonstration boats such as Herreshoff built, but quite impracticable in commercial boats, because the propeller cannot be used below the heel of the stern frame. In regard to twin-screw shafting, if you keep the propellers apart to get the additional efficiency by more solid water you may expose the propellers to possible damage. But before you can arrange these shaft lines you must design the ship, and when you have arranged your shaft lines to get efficiency from the propellers you should arrange your stern and counter adequately to protect them. In reply to the question of water drip, and the use of distilled water. I hope, and we all hope, that water drip will soon be eliminated. Most manufacturers are working to that end, and some have obtained greater success than others; and some claim greater success than they have attained. There is no doubt that in a few years water drip will be entirely dispensed with, and any engine working with water drip will be quite out of the market.

SESSION 1920-21.

President: LORD WEIR OF EASTWOOD.

Scheme of Operation in Petrol and Semi-Diesel Type Engines, Forms of Scavenging, Operation of Valves, Fuel Pumps and Ignition.

BY A. W. BRADBURY (Member).

Read, Tuesday, January 18, 1921.

IN the reciprocating steam engine, power is derived by the admission of steam under considerable pressure into a cylinder behind a piston, which is driven outward by expansion of the steam, and, through a connecting rod and a crank, this piston thrust rotates a shaft connected direct or by belt or gears to the object to be moved. In this process, no chemical action takes place within the steam cylinder. All chemical action occurring in the generation of steam power is carried out in the fire under the boiler and the power stored in the boiler in the form of steam under pressure. The action of the steam engine itself, in using this stored power, is purely mechanical.

In the internal combustion engine both mechanical and chemical action take place, wholly within the cylinder. Creation of pressure, heat and expansion, in this order, all take place within the cylinder behind the piston and drive it outward or downward just as steam drives a piston.

In both cases the fuel consists of carbon and hydrogen, principally the former, mixed with the oxygen of the air in such proportion as to support combustion of the fuel elements. In the generation of steam power, coal, coke and wood are convenient fuels and are the fuels generally employed, but liquid or gaseous fuels may be used, and are used. But this process is wasteful because the quantity of oil, alcohol or gas burned under a boiler to generate a given quantity of power in a steam engine, will supply more than twice that quantity of power if consumed in an internal combustion engine. Coal cannot well be employed as fuel for an internal combustion engine unless first finely pulverised, hence the fuels for these engines are found in the liquid or gaseous form. If the fuel be a gas it is mixed with air and drawn into the engine by the suction of the piston, the mixture is compressed in the cylinder by the inward stroke of the piston and then ignited by an electric spark. Combustion of the mixture is practically instantaneous, the resulting heat causes rapid and intense expansion of the contents of the cylinder and this expansion forces the piston outward on its power stroke.

If the fuel be a light liquid, that is to say, of low specific gravity, so that it can be gasified by passing air rapidly over it or through it, it is led into a carburettor so arranged that a quantity of fuel just sufficient for each charge is caught in an air current created by piston suction and drawn into the cylinder, compressed and fired, as in the case of a purely gaseous fuel. The various spirits of petroleum known as petrol and a distillate of coal called benzole are the fuels generally employed by means of carburation. Certain heavier grades of petroleum oils, known as paraffins, cannot be carburetted like petrol or benzole, but must be vaporised by the application of heat before they can be mixed with air in such proportions as to burn readily and cleanly in a cylinder. Such vaporisation is usually accomplished in a vaporiser surrounding the exhaust pipe and heated by exhaust gases, after having obtained the initial heat either artificially or by running the engine on petrol for a few minutes.

The still heavier fractions of petroleum cannot be vaporised and burned in the same manner as paraffin without the accumulation of deposits of carbon and gum on the piston and in the cylinder, to the detriment of the engine. These fuels are forced directly into the engine by pump or by a blast of compressed air, through an atomiser which breaks them up into very fine globules. These globules, surrounded by air rapidly increasing in temperature through the heat of compression, are converted into a vapour or gas, and then burned the same as are the gases of the lighter fuels. Whatever the form of fuel may be, or whatever process is pursued in preparing it for combustion and mixing it with air in proper proportion, the final result is the same in producing pressure on the piston and driving it outward.

At or near the end of the outward stroke of the piston the nearly spent gases are allowed to escape from the cylinder, through an exhaust valve or a row of ports, and a new mixture of air and fuel, or air only, is taken into the cylinder to be compressed ready for the next power stroke. The "Scheme of operation" of an internal combustion engine is one of "alternate expansion and contraction"—expansion being caused by heating and contraction by cooling the contents of the cylinder, such cooling taking place very rapidly upon release of pressure on the gases by the opening of the exhaust outlet. The compression of the charge also constitutes a form of forced contraction, the force for which is supplied by the momentum of the flywheel in engines of one cylinder and by expansion in one cylinder against compression in another in multi-cylinder engines.

One expansion stroke and one compression stroke together, constitute a cycle in the two-cycle type of engine. In this form every outward stroke of the piston is a power stroke, every inward movement a compression stroke. Here the burnt gas must escape and the fresh charge of air and fuel must enter the cylinder while the piston is at or near the outward limit of its traverse. This necessitates a rapid emptying and refilling of the cylinder. In engine parlance, the process of ridding the cylinder of the products of combustion, *i.e.*, steam, carbon dioxide and other inert gases, and replacing them with fresh air or air and fuel mixture, is termed scavenging the cylinder. In a two-stroke engine the cylinder is scavenged with every revolution of the flywheel.

In this type of engine, scavenging is attained usually by two rows of ports in the cylinder barrel, placed opposite each other, the ports on the exhaust side being higher, or deeper than those on the inlet side. During the greater part of the stroke these ports are covered, and therefore closed, by the piston. As it moves outward on the power stroke the piston first uncovers the upper or inner portion of the exhaust ports and the burnt gases above the piston, still under pressure, begin to rush out. With the sudden release from pressure the exhaust gas contracts enormously. While it is rushing out, with the piston moving all the time and increasing the area of opening of the exhaust, the piston begins to uncover and continues to enlarge the inlet openings, whereupon the fresh charge of air, or air and fuel, flows in through these. The escape of the burnt gas is continued through the momentum set up in the mass before the uncovering of the inlet ports, and this momentum would cause a partial vacuum in the cylinder if the inlet ports were kept closed. The entrance of the fresh charge is induced partly by the fall of pressure within the cylinder and partly because this charge itself is under pressure, for it has been taken into the crankcase during the inward movement of the piston, and there compressed to a few pounds above atmosphere by the outward movement, the piston thus acting as a pump attached to the crankcase. A by-pass connects the crankcase with the inlet ports, through which the fresh charge flows to the latter. If nothing were done to prevent it the incoming fresh charge probably would flow straight across the top of the piston and out through the exhaust ports. To obviate this, a baffle ridge or plate is formed on the top of the piston, so shaped that it directs the inflowing charge toward the top of the cylinder.

Thus we find in the cylinder at the same instant, two currents flowing in opposite directions, one composed of incombustible gas flowing downward and outward, the other of air or combustible mixture flowing inward and upward.

Is it possible for these currents to pass each other without some intermingling of the two? Evidently not. If there be intermingling of the two, either some of the fresh charge must escape with the exhaust gas or a portion of the latter must remain in the cylinder to dilute the new charge, or both. Now, no cylinder is ever completely scavenged of exhaust gas in any engine, because the volume swept by the piston is less than the total volume occupied by the piston, plus the clearance space above it. Suppose the volume swept by the piston to be 100 cubic inches and the clearance volume above the piston when at the top of its stroke to be 10 cubic inches. The total is 110 cubic inches. The combustible charge is compressed by the piston into this 10 cubic inches of clearance space and fired. The piston then moves outward and increases the total volume by 100 cubic inches. At the same time 100 cubic inches of air or mixture has been compressed in the crankcase, and this flows into the cylinder to take the place of the 100 cubic inches passed out through the exhaust ports. Ten cubic inches of burnt gas remains in the cylinder to mingle with the new charge and dilute it to that extent, provided the two charges have not intermingled in passing each other, the one on the way out through the exhaust ports, the other on the way to the top of the cylinder. Since it appears impossible to keep them entirely separated, a small portion at least of the incoming charge must escape. And if a portion of this charge escapes, a corresponding portion of the burnt charge must remain in the cylinder. This portion, added to the 10 cubic inches already retained, increases the dilution and reduces the power of the following explosion below what it would be if the charge were diluted only 10 per cent. Hence we conclude that the ordinary two-stroke engine is not scavenged as it should be, and this imperfect scavenging probably accounts for the fact that a two-stroke engine of given bore and stroke does not develop twice the power of a four-stroke engine of the same bore, stroke and speed of revolution, but with half the number of power impulses per minute.

Various schemes have been put forward for improving the scavenging of two-stroke cycle engines, in which the two rows of ports have been retained in the same end of the cylinder. Only one of these schemes appears to have proved of sufficient value to justify its retention in view of the increased cost of an engine so fitted. This consists of an auxiliary row of inlet ports, situated above the regular ports, through which additional air can be forced into the cylinder after the exhaust ports have been closed. This plan does increase the volumetric efficiency of the engine. But how can it improve scavenging *per se*?

In that form of engine in which two opposed pistons work in the same cylinder barrel, with the charge compressed and fired between the two and driving the pistons away from each other, scavenging is improved by the placing of a row of inlet ports at one end of the cylinder barrel and the exhaust ports at the other, both rows of ports extending clear around the barrel. Also in one type of large gas engine where two parallel cylinders are connected by a passage at the top, and the two pistons move in unison, with a row of inlet ports clear around the bottom of one cylinder and exhaust ports around the bottom of the other, both cylinders are swept as clean at every stroke as any four-stroke engine could be. In neither of these arrangements do we have two currents of gases moving in opposite directions in the same cylinder at the same time.

The period of time during which scavenging must take place in a two-stroke engine is very short. The general rule is to make the area of the exhaust ports such that they allow for a gas movement of forty-five feet per second, based on piston volume and speed of rotation, and the inlet area large enough to permit the air to pass at eighty feet per second. If we follow these rules in an engine running at three hundred revolutions per minute, or five revolutions per second, and if the exhaust ports be open during one hundred and twenty degrees of the crank path, the gas must escape in one-third of one-fifth of a second, that is, one-fifteenth of a second, or fifteen times forty-five feet per second, which equals 675 feet per second. The same ratio of port area to piston volume and speed should be maintained in other sizes running at other speeds, and such areas are ample. But how about the smaller inlet ports? They are shorter, measured in the direction of the stroke, even if they extend through a longer arc. They must be shorter if the piston is to uncover the inlet after the exhaust and close the inlet before the exhaust is closed. Usually the inlet ports are a little more than half as high as the exhaust ports. Continuing the foregoing illustration, if the exhaust ports are open through 120 degrees of the crank path, the inlet ports will be open through, say, ninety degrees, or one-fourth of a revolution. If one revolution occur in one-fifth of a second, the air must enter in one-fourth of one-fifth, or one-twentieth of a second, *i.e.*, flowing at 20 times 80 feet, or 1,600 feet per second.

If the rate of flow of the ingoing charge be nearly three times the rate of discharge through the exhaust, one may be inclined to speculate as to whether, after all, this rapid movement does not account for the scavenging of the cylinder. Whether the entering charge does not, indeed, "expel the burnt gas," an expression we frequently hear or read. It might be so if all the force exerted were pressure on the inflow. But it is not. The contraction and escape of the burnt gas, together with its *momentum*, tend to create a vacuum in the cylinder which exerts a pull on the entering charge greatly in excess of the four or five pounds per square inch exerted by crankcase pressure.

A speed of 1,600 feet per second for air or gas in an engine seems pretty high. This equals 96,000 feet per minute, or over eighteen miles. It means a hurricane of 1,080 miles per hour—equalling the speed of the earth's surface at the equator. Such speeds, and even higher speeds, occur regularly in the gas and air movements of a two-stroke engine, and the only way to lessen them is to make the ports larger. Conditions limit the size of the ports, however. Where both rows are located in the same plane it is evident that each cannot extend half-way round the cylinder without increasing the tendency for exhaust gases and fresh charges to intermingle. Whilst, if the ports be made high, the exhaust gas is released while still under considerable pressure, and power thus wasted. One way out of the difficulty would be to locate the ports in different planes, but this cannot be attained in the same end of the cylinder barrel without shaping the top of the piston in such a way that the expanding gases will exert a side pressure on it.

The latest attempt to solve the scavenging problem in the two-stroke engine consists in a uniflow arrangement in a single acting cylinder, in which the top of the piston is perfectly flat and the exhaust ports extend clear around the cylinder barrel, maintaining the pressure on the piston throughout nine-tenths of its stroke, and the fresh charge is admitted at the other end of the barrel. Theoretically this should increase the power and should be more economical of fuel than is, say, a submarine engine in which the exhaust ports are uncovered when the piston is only three-quarters of the way out on its stroke.

In the four-stroke cycle engine we have exhaust gases escaping, and a fresh charge entering the same end of the cylinder, but not at the same time. Instead of completing the cycle in two-strokes, we have here two inward and two outward movements of the piston in which to complete it. This system enables the cylinder to be properly scavenged without any intermingling of opposing currents. Here the piston plays the part of a pump within the cylinder itself, and the crankcase has nothing to do with the cycle. Beginning with a power impulse, the piston moves outward. At or near the end of the outward stroke a valve opens in the top of the cylinder and the returning piston pushes the burnt gas out through this valve. About the time the piston returns to the starting point this exhaust valve closes, and an inlet valve opens, which is in communication with air, or air and fuel. The piston then moves on its second outward stroke and draws air and fuel through the inlet valve into the cylinder. At the end of this stroke the inlet valve closes. Both valves being now closed, with the fresh charge imprisoned in the cylinder, the second inward stroke of the piston compresses the charge, which then ignites and starts the piston outward on a new cycle. Scavenging is not so hurried here as it is in the two-stroke engine. One whole inward movement of the piston is given to forcing exhaust gases out of the cylinder. One whole outward movement to inducting a fresh charge into the cylinder. Thus the four-stroke engine spends half of its time in compressing, firing and expanding the power charge, and the other half in pumping out burnt gas and pumping in air and fuel. It gives us one power impulse to every four-strokes of the piston. The other type, one power impulse to every two strokes of the piston. If the two types run at the same number of revolutions per minute, the two-stroke engine fires twice as many charges of fuel in the same period of time as does the four-stroke. If the pressure on the piston were carried as far out in a two-stroke engine, and if scavenging were as thorough, we would expect this type to develop twice the power of a four-stroke having the same cylinder capacity and the same speed of rotation. But it doesn't do it.

As to scavenging, we will revert to our previous example for the purpose of comparison. If the volume swept by the piston of a four-stroke engine be 100 cubic inches and the clearance space 10 cubic inches, the piston will expel 100 cubic inches of burnt gas and only 10 cubic inches will remain in the cylinder. In fact, with valve openings of ample area and valve action properly timed, it is possible to expel more than 100 cubic inches from the cylinder in question. In doing this the momentum of the gas plays a part, for it has weight. Once set moving it has a tendency to continue that movement until its momentum is expended. If thirteen cubic feet of air at sea level weigh one pound, one cubic foot will weigh one-thirteenth of a pound. Hot gas will weigh less. For the purpose of illustration, let us say one cubic foot of gas weighs one ounce. Get this moving at 600 feet or 1,000 feet or 1,500 feet per second and it will tend to continue moving in the same direction after the starting force is withdrawn. Now if the exhaust valve be kept open until the momentum of the gas is expended and then suddenly closed, it is possible thus to create a partial vacuum within the cylinder. In other words, more than 100 cubic inches of gas will have been removed from the cylinder and, therefore, less than 10 cubic inches will remain in it. Now open the inlet valve with the piston already started on the suction stroke, keep this valve open beyond the bottom centre, and perhaps we can take in fresh charge enough not only to fill the cylinder to atmospheric pressure but a little more, due to the momentum again. This actually occurs in four-stroke engines of high speed. "High speed" does not necessarily mean a great number of revolutions per minute, but it means rapid piston travel. Thus, if the stroke be 12 inches the piston will travel two feet per revolution. If such an engine run at 600 revolutions per minute, the piston will travel 1,200 feet per minute. It is evident that the faster the piston travels the greater the momentum it will set up in the expelled gases and the faster the air will rush in on the suction stroke, thus increasing its momentum in the inward direction.

In timing the valves of a four-stroke engine these facts are taken into consideration. If the valves were opened and closed just at the end of the piston stroke they would be open through just 180 degrees of the crank path. They would then be open through a period considerably longer than is possible in a two-stroke engine. But they are open for more than 180 degrees of crank angle in the four-stroke, because it is necessary, particularly in high speed engines, to give some lead. This is to permit air and gases to get under way. A brief period of time is required for air or gas to start moving just as is required for them to stop moving. In present-day four-stroke Diesel engine practice, the valve settings are usually as follows:—The exhaust valve opens at 35 degrees before bottom centre and closes five degrees after top centre, thus being open through 220 degrees of crank path. The inlet valves open at 10 degrees after top centre and close 30 degrees after bottom centre, thus being open through 200 degrees of crank path. At the top of the stroke, then, there is an interval of five degrees between the closing of the exhaust and the opening of the inlet valve. This setting causes compression of the charge through 150 degrees and expansion on the power stroke through 145 degrees of crank path. Valve settings vary somewhat from the foregoing, particularly in small engines. For instance, in high speed petrol engines, with very light reciprocating parts, we may find the exhaust open through 230 degrees and the inlet through 210 degrees of crank angle, sometimes with an overlap instead of an interval at the top of the stroke.

Thus far we have dealt only with two-stroke engines in which the scavenging is accomplished through ports only, and with four-stroke engines in which scavenging takes place through mechanically operated valves. We now come to a modified form which is a two-stroke engine, so far as the cycle is concerned, but which employs mechanically operated valves in the scavenging. In this form the usual exhaust ports are found, and they function in the usual manner. In addition there is placed in the cylinder head a scavenging valve, mechanically operated and communicating with a supply of air under pressure. On the power stroke of the piston, shortly after the upper edge or the exhaust ports is uncovered, the air valve is opened and the compressed air blows the exhaust gases out. This valve then closes before the piston has fully covered the exhaust ports again. This arrangement assures satisfactory scavenging, but at the cost of an air pump with its connections, in addition to the cost of the valve and its operating mechanism. Still, in large engines, where the added cost bears small relation to total outlay on the plant, this method of scavenging has been found to be commercially practical. When the same results in scavenging can be attained in all types of two-stroke engines, large and small, without the extra air pump and valve operating mechanism, we shall have made a step forward in two-stroke design which will be of greater value than any other improvement made in this type in many years. For, undoubtedly, the weakest feature of the two-stroke heretofore has been in the scavenging of the cylinder. Such an improvement can be made, and in fact, has been made. An engine embodying it will shortly appear in the field. With the scavenging bugbear removed, the prejudice against crankcase compression still remains, at least in the minds of those who have not had extensive practical experience with the actual working of two-stroke engines. The cry of the uneducated is that you cannot see what is happening to the big end when closed in out of sight. The answer is that nothing happens to it. Another answer is that the big end is not throwing oil all over the place when closed in. But the man with the big phrenological bump of prejudice says the main bearings throw oil which the compressed air in the crankcase forces outward. The answer to this is that they don't, if very simple precautions be taken to prevent it. Another complaint is that the automatic air inlet valves in the crankcase cause trouble. Wrong again; if they are properly made, and not all two-stroke engines have them. Some have a third port uncovered by the piston when near the top of its stroke to admit air to the crankcase. Either arrangement is satisfactory.

The three port, crankcase compression, two-stroke engine is valveless. It is the simplest engine extant. Simpler than the simplest steam engine. It originated in England as a petrol engine. The inventor of the two-stroke petrol engine issued licences to firms in foreign countries, one of which built and sold over sixty thousand two-stroke engines prior to 1912.

Two-stroke engines have been in operation in every country in the world for more than two decades, the total number long ago exceeding the million mark. While such large numbers of them have been finding their way into every corner of the earth, British builders have stuck to the four-stroke type. They have not expended money or energy in the development of the two-stroke. Had they done so, there would to-day be less prejudice against this type. Power for power, the two-stroke is cheaper to build, lighter in weight, occupies less space and requires fewer repairs than a four-stroke. The popularity of the semi-Diesel and the success of the larger Diesels operating on the two-stroke cycle are gradually bringing the two-stroke engine into its own.

It requires no valves, no valve springs, no tappets, no cams, no cam shaft, no half-time gear, no cam shaft bearings, no valve cages. Deduct the expense of these from your four-stroke job and see the difference in first cost. Or add the cost of these to a two-stroke engine and see how much more it costs. Then in running, you have no valves to grind, no tappets to adjust, no gears to set to time, no cam shaft bearings to oil, none of these things to go wrong. In the four-stroke you have just two items to set against these advantages in the two-stroke, viz., a slightly lower fuel consumption and slightly better flexibility. With improved scavenging the difference in fuel consumption will become practically nil and the difference in flexibility is not a vital point. If the two-stroke cycle engine were not satisfactory to those who use them to earn a living, we should not find twenty semi-Diesels of the two-stroke type to one of the four-stroke. And unless those who devote their lives to the design of engines were not satisfied that the two-stroke is perfectly practical from every standpoint, we should not see one large two-stroke Diesel after another rapidly coming into competition with the already established four-stroke machines.

If cams be employed in a two-stroke engine they will, of course, be worked from the crank shaft or from a cam shaft running at the same speed as the crank shaft. In the four-stroke engine they must be worked from a cam shaft running at half the speed of the crank shaft. The most common arrangement is to mount the cam shaft parallel with the crank shaft and actuate it through a pair of two to one or half-time spur gears, with the smaller gear keyed to the crank shaft, the larger to the cam shaft. There is a cam for the exhaust valve and another for the inlet valve. They are so set on the shaft that the one will lift the exhaust valve during the exhaust stroke and the other raise the inlet valve during the suction stroke of the piston, while both valves are kept closed during the compression and expansion strokes. The cams should be made of hard steel and the tappet roller should run on the cam all the way around, so that the cam cannot knock against the roller. A slight clearance must be left between tappet and valve stem, when the valve is closed, but only enough to make sure that the valve does close tightly and is not held off its seat when the roller is running between the two ends of the cam rise. The length of the cam rise in degrees is half the length of the arc of the crank path through which the crank travels while the valve is to be held open, because the crank shaft is rotating at twice the speed of the cam shaft. Thus 100 degrees of cam shaft angle is equivalent to 200 degrees of crank shaft angle.

In calculating valve port areas, allowance must be made for the area occupied by the valve stem, and the port openings should be large enough to permit the passage of air and gas at 45 to 50 feet per second, with a valve lift equivalent to one-fourth of the diameter of the port opening. Valves and stems should be made in one piece, of nickel steel, and exhaust valves particularly should be well cooled by liberal water jackets.

In gas, petrol and paraffin engines, where the fuel is mixed with air before admission to the cylinder, throttling is accomplished simply by a reduction of the quantity of mixture taken in on the suction stroke, by a butterfly valve placed in the induction pipe. This valve controls the quantity of air admitted, and since the quantity of fuel entering the cylinder depends upon the air current, the richness of the mixture is not greatly affected by the action of the throttle valve. And because the charge is fired by an electric spark, ignition is certain.

But in an engine in which air alone is drawn into the cylinder to be compressed and the fuel forced into the air charge when compression is far advanced, the case becomes entirely different. In the Diesel engine where we depend solely upon the heat of compression to fire the charge, it is evident that to throttle the charge of air would be to reduce the compression, lose that heat and fail to fire.

In the semi-Diesel engine, ignition is brought about by a "hot spot," which depends upon obtaining its heat partly from compression. True, it is the burning of the fuel which raises the temperature of the hot spot to redness, but because it is already warmed to a considerable degree by the heat of compression, the explosion heat carries it on up to a temperature at which it ignites the fuel. Take away the heat of compression and the hot spot sooner or later cools down to a point where it will not fire the charge, and this is what happens when we reduce compression by throttling the air inlet.

In this type of engine the compression of the air varies more than in any other type. In some semi-Diesels, compression is as low as nine atmospheres, isothermal. In others, it is as high as eighteen atmospheres. In two or three engines it is still higher. The average semi-Diesel compression is perhaps in the neighbourhood of 150 pounds per square inch isothermal or, say, 210 pounds adiabatic, if we disregard the effect of the cooling water. Owing to the rapid compression, and the rapid repetition of compressions in running on full load, the cooling water probably does not affect the adiabatic temperature to any marked degree. If the air be uncooled, rapid compression of it to 150 pounds per square inch raises it to a temperature of 589 degrees Fahrenheit, if taken into the compressor at a temperature of 60 degrees F. Such a temperature is high enough to vaporise any of the mineral oils used as fuel in an oil engine. Presumably such compression also raises the temperature of the hot spot to a corresponding degree. The design of the engine is such that the heat of combustion will then add to the hot spot the heat required to raise it to the point where it ignites the oil, and maintain it at that temperature, even when the engine is running light. It is obvious that if we throttle the air inlet in an engine of this type, the heat of compression will be very much reduced. If we reduce compression to 100 pounds per square inch we obtain an adiabatic temperature of 485 degrees Fahrenheit. In reducing the pressure fifty pounds we have reduced the temperature 104 degrees. If the hot spot be reduced 100 degrees in temperature, irregular firing is sure to follow, and soon after it will fail entirely.

In order to run on part load or no load it is better to go on compressing the full quantity of air at every revolution, and thus retain all the heat of compression, and to regulate the speed of the engine by reducing the amount of fuel fed. We pay no attention to mixtures, "rich" or "lean," in the semi-Diesel engine, nor in the true Diesel, for that matter. There is always in these engines a quantity of air in excess of that required for combustion, so that, in governing them, we are not obliged to reduce the air when fuel is reduced, nor to increase the quantity of air to meet an overload. Another point to be remembered is, that all power expended in compressing air in the cylinder is returned in the rebound of the compressed, and therefore heated, mass. In a few makes of semi-Diesel engines an arrangement is fitted for throttling the air in the by-pass between the crankcase and the cylinder. On the other hand an engine brought out recently has a valve which prevents air from entering the crankcase until the piston is part way on its inward movement, and has created a partial vacuum in the crank-case. The valve then opens, permitting the air to enter with a rush, which continues until the air in the crankcase exceeds atmospheric pressure, upon which the valve closes and the return of the piston compresses this air ready for scavenging the cylinder. The same effect could be produced in the ordinary three port engine, possibly, by increasing the area of the crankcase inlet ports circumferentially. The effort to get air into the crankcase above atmospheric pressure certainly seems more reasonable than to reduce the scavenging air by throttling it. Not less scavenging air, but *more* is what we want in the ordinary, two-stroke semi-Diesel engine, in which both the exhaust and inlet ports are in the same end of the cylinder barrel and are both open at the same time. In this type we do not lose any fuel in scavenging, as may occur in those engines where the air and fuel *mixture* is employed for scavenging. If we can get enough scavenging air through the semi-Diesel cylinder to sweep out *all* of the burnt gas, we shall then compress nothing but pure air. We can then burn more fuel than we do where a certain quantity of burnt gas remains mixed with the fresh air charge. As for indicator cards, they are useful in studying all the idiosyncrasies of an engine of new design. If we wish to observe scientifically and record every item of interest in a new engine we must use an indicator. Yet we obtain data occasionally for which we cannot fully account. For general use any good brake, fuel consumption, revolutions per minute and cooling water used will give us all the data we need to determine whether an engine is good, bad or indifferent. This subject recalls a scientific test made on an oil engine some years ago by a college professor, who neglected nothing. He had pressure gauges, indicators, revolution counters, chronometers, thermometers, barometers, water meters, calorimeters, and something to measure the smoke. Among other items in a lengthy report he stated that the water used for cooling was eleven gallons per horse power per hour and that the mean effective pressure of the engine was twenty-nine pounds per square inch. Now if it had only occurred to the professor to use 29 gallons of cooling water per horse power hour he probably would have gotten a mean effective pressure of 11 pounds per square inch! This story may serve to emphasise the fact that these engines should run warm. The cooling water should leave the engine at a high temperature, say from 180 degrees to 200 degrees Fahrenheit. A good plan is to have the cooling water pass through the jacket of the exhaust manifold on its way to the cylinder jackets.

Under the head of Fuel Injection Valves the framers of the Syllabus evidently meant to include fuel injection nozzles and fuel pumps, because the entire system of fuel feed is so intimately connected, and because, whether a fuel nozzle contains a valve or not, it must be supplied with a pump which measures and delivers the oil to the nozzle. The prime object of any fuel injection system is to break up the fuel oil into a fine mist—the finer the better up to a certain limit—and to spread this mist evenly throughout the charge of air with which it is to combine in burning.

Fuel injection valves are employed in all makes of Diesel engines and in a large proportion of the semi-Diesel type. In the Diesel engine the injection valve is arranged to retain within the nozzle the fuel required for each charge until the proper instant for injection, then to lift and permit the fuel to be blown into the cylinder by an air blast under a pressure of 900 to 1,000 pounds per square inch, through a series of staggered holes in thin steel plates fixed in the fuel nozzle. In this case the fuel pump exerts no pressure save that necessary to deliver the oil into the nozzle space surrounding the injection valve. The latter when closed prevents both air and oil from entering the cylinder. At the proper point in the cycle, the valve is opened mechanically and the air blast effects the injection. In such case the fuel nozzle possesses a single, central opening and the valve is tapered to a needle point, to plug the opening when the valve is closed. In Diesel engines having a so-called solid injection the arrangement is different. Here no air blast is used. Instead of air, a resilient fuel pipe is employed and the fuel forced into this by a ram pump to a very high pressure. This fuel supply pipe is made of thin, high tensile steel, is oval in section, and extends along the side of the cylinders, with a small branch pipe leading to each injection valve. The tips of the fuel nozzles are hemispherical in shape and each contains several very small holes, which diverge somewhat from the centre line, and communicate with a central hole, which is plugged by the fuel valve. The pumping of fuel into the oval pipe to a pressure of something like four thousand, and in some cases even as high as 8,000 pounds per square inch, causes this pipe to spring outward in the direction of its shorter diameter, that is, the high pressure tends to spring its flat sides outward and make the section more nearly round. Then when a fuel valve opens and thus relieves the pressure somewhat, the return of the pipe toward its original shape is the force which injects the oil into the cylinder.

In the semi-Diesel engine, lower fuel pressures are involved because the compression pressures in the cylinders are not so high as in the true Diesel. The greater number of semi-Diesels have fuel nozzles in which there is a single central spray hole. Just a plain round hole in the smaller sizes and a hole with rifles in it for the larger cylinders. In calculating the size of this hole the volume swept by the piston is the basis upon which to work, regardless of the speed of the engine. For example, a swept volume of 300 cubic inches would require a spray hole about fifteen thousandths of an inch in diameter. The shorter the hole, that is, the thinner the metal through which it is drilled, the better the atomisation of the oil, for a long hole through thick metal tends to wiredraw the oil and feed it in a thin stream. In some engines having fuel nozzles of this type, a simple check valve is placed as near to the point of the nozzle as possible, while in others no valve is put in the nozzle. In the latter a short, sharp stroke of the fuel pump is relied upon to break up the oil at the spray hole. Other engines have a needle valve, spring loaded, which is opened by the pressure of the fuel oil, and this is better, because less likely to permit any dribble of the oil at the finish of the injection. In the early days of the semi-Diesel, or hot bulb engine, some of the injection nozzles were fearfully and wonderfully made. An instance may be cited in which the fuel nozzle was a plain, steel tube projecting into the combustion chamber some two inches or more. This pipe was plugged at the point and had the little spray hole drilled in the upper side near the inner end. The result was that about every six hours this spray hole became filled with carbon and the engine stopped. The makers met this by supplying half a dozen spare nozzles, with instruction to remove the stopped-up nozzle and replace it with a clean one, as often as required. But those who ran the engines complained of the stoppages and of the difficulty of freeing a pin-point hole of tough coke and also of the fact that frequent cleanings soon enlarged the hole so much that it failed to atomise the oil. Then the makers cut a small notch in the point of the fuel nozzle and screwed the plug in so that the flange of the plug formed the third side of a triangular hole, and instructed operators to decarbonise this by removing the plug and cleaning the notch with the thumb nail. Surely the nail would not damage the hole! But this carbonised worse than the round hole and nozzles had to be cleaned every three hours instead of six. Upon examination of the nozzles a coating of carbon or coke was found lining the whole interior. This could be removed only by thorough soaking and scrubbing with paraffin. If not entirely removed, particles of it soon broke off and plugged the spray hole. Imagine an uncooled tube five sixteenths of an inch inside diameter filled with paraffin, projecting into a combustion chamber with temperatures rising to 2,700 degrees Fahrenheit and higher at every explosion, and are we surprised that it becomes filled with carbon? Whatever the form of spray hole and whether valveless or not, the fuel nozzle should always be well water cooled. And unless it contains a valve it should always be inclined upward at a considerable angle above the horizontal, to prevent air locks and to guard against dribbling of the fuel at the end of the injection. An air bubble in the fuel line anywhere between the pump and the nozzle is fatal to regular running of the engine. For this reason every fuel nozzle must be designed and mounted in such a way that air locks cannot possible occur.

If injection nozzles and valves be properly designed and carefully made; if they be thoroughly water cooled; if they be mounted properly, and if their connections are all they should be, they are not likely to give any trouble except from *dirty fuel*. This is the cause of most of the trouble arising in this particular fitting. It is essential, then, that fuel filters be ample in number and made of gauze fine enough to strain out all particles of dirt large enough to choke the spray hole or interfere with the seating properly of the injection valve. If the mesh of the gauze be smaller than the spray holes, and if strainers be cleaned frequently, no dirt is liable to reach the spray holes. And yet, when an engines stops suddenly without apparent cause the first rule is, *look for dirt*.

There is a type of engine, known on the Continent of Europe as the Brons and on the Western Continent under half a dozen different names, in which no force is required to feed the fuel.

Hence the function of the injection valve is solely to measure the charge. In this type of engine a small steel cup is placed in the combustion chamber, having a number of fine perforations in the sides. On the suction stroke of the engine the fuel is deposited in this cup, and on the return stroke of the piston it is vaporised and ignited by the heat of compression. This, of course, involves compression pressures practically as high as those found in the Diesel engine.

As to what actually takes place in the Brons cylinder, the theory is that the rising heat of compression first vaporises a small quantity of fuel in the cup, then ignites it, and the tiny explosion following blows the balance of the fuel out through the fine holes in the cup, into the cylinder where it burns. We may not be able to agree with this theory, but the fact remains that the Brons engine works beautifully, is economical of fuel, burns any grade of fuel that the Diesel does and is much simpler in construction than the latter. In view of these advantages one is at a loss to understand why British manufacturers have not given more attention to this type.

Fuel pumps of plunger pattern are of course necessary for supplying fuel under pressure to any type of oil engine. They are of two varieties, one of which has a suction valve on one side of the single plunger and a discharge valve on the other. Where balls are used as valves we sometimes find two on each side of the plunger—an indication that the designer does not have full faith in a single ball valve. And he should not have. The strongest argument in favour of a ball valve is that it is cheap. The other variety of fuel pump has two plungers, the secondary one of which is controlled by the governor in such a way as to keep the suction valve open for a shorter or a longer period, according to the quantity of fuel required to correspond with the load on the engine. With the single plunger pump, governing is secured by varying the length of stroke of the plunger. Whatever the style of pump, it should be made of proper materials and the workmanship should be as nearly perfect as it is possible to attain, for the fuel pump is the heart of the engine. As to whether fuel pumps should be packed, those of wide experience hold opposing views on the question. A pump properly packed should work for years without leakage. But few makers, and still fewer operators, seem to be able to pack a fuel pump properly. The strongest argument against packing is that, although it may leave the works in perfect condition, somebody in the wilds of Africa or South America is likely to pull the plunger out, find he cannot replace it, then remove the packing and try to repack. He may have received printed instructions as to packing a fuel pump, but he cannot or does not follow them, his pump leaks when he remounts it and trouble haunts him thereafter. If a pump be made tight under oil pressure, without packing, it will work as long, without leaking, as will a packed pump, while the plunger may be removed and replaced at pleasure without risk of rendering the pump inefficient. Fuel pumps can be made, and are made, which work under pressures of several hundred pounds per square inch for long periods without showing any sign of leakage and without any packing whatever.

Small oil engines, up to about fifteen horse power, are started by hand, by pulling the flywheel against compression. These cannot be started by revolving them in the direction of running, whether they be two-stroke or four-stroke, because ignition cannot be retarded, as is done in an electric fired engine. In pulling against compression the charge fires before the piston reaches top centre, and starts the engine back the other way.

Stationary engines having large flywheels with spokes require no other device for starting them. Using the spokes as handles, the flywheel is thrown against compression, that is to say, it is given as much momentum as possible and the operator lets go just before the engine fires. For if he holds on until the charge ignites he may receive a sprained wrist.

In marine engines having flywheels of disc pattern the best provision for starting by hand is the fitting of a disappearing handle in the rim of the wheel. By hooking three fingers over this handle and pulling against compression until the firing of the charge jerks the handle away from the fingers, the engine starts at the first pull and the operator runs no possible risk of injury, for the handle flies in flush with the wheel the instant it is freed from the hand. Such a handle is useful also for turning the engine over, in sizes too large to be started by hand. In still larger sizes, where more leverage is necessary to turn the engine, holes should be provided in the face of the wheel rim into which a bar may be inserted, but these holes should not extend entirely through the rim. If they do, there is danger of shoving the bar too far through and failing to get it out quick enough if the engine fires. This bar is not to be used for starting the engine, but only for turning into position for starting,

SCHEME OF OPERATION IN PETROL AND SEMI-DIESEL TYPE ENGINES.

but sometimes when turning an engine over while hot, it will fire unexpectedly. The cranking bar should be made specially for the purpose, straight on one side and rounded longitudinally on the other, so that only the straight or flat side can bear against the side of the hole. It should have a projection or hook on the rounded side to bear against the face of the wheel. If the wheel should then move suddenly in the direction it is being turned, the bar will be forced out of the hole. Relief valves in cylinders should always be opened before trying to turn a warm engine.

In engines too large to be started by hand, compressed gas or compressed air is generally employed The gas is accumulated while the engine is running, through a special valve fitted in the cylinder and connected to the starting tank. The objections to this system are: 1st, that the connecting pipe gets hot; 2nd, the valve stems sometimes stick, owing to the heat and to the impossibility of lubricating them; 3rd, carbon deposits form on the valves of the connections, causing loss of compressed gas by leakage. For these reasons compressed air is preferable, as it has none of these drawbacks. Accumulation of the air is best obtained by means of a small air compressor attached to the engine. This need not be large enough to require water cooling. It runs while the engine is running and stores air enough for starting purposes. It should be fitted with a relief valve or,

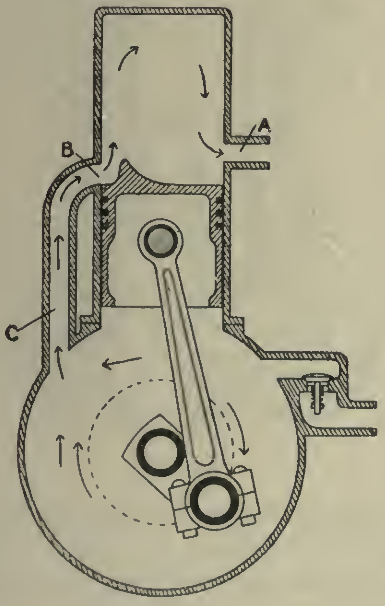


Fig. 1.

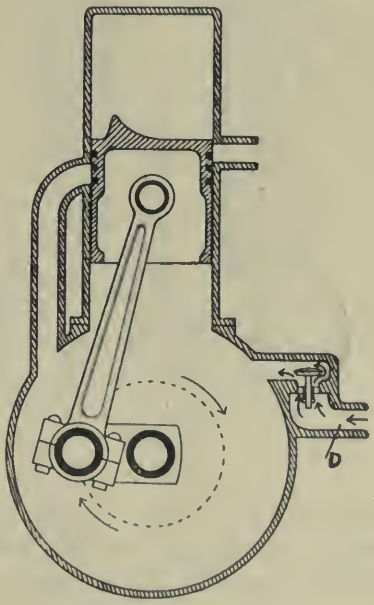


Fig. 2.

better, with a three way cock, so that it can be relieved from compressing air when the pressure in the tank is as high as required. Of course, neither gas nor air can be accumulated by an engine for starting the first time, nor for starting it in the event of the pressure tank having been emptied by a leak. This makes necessary an auxiliary arrangement of some kind and

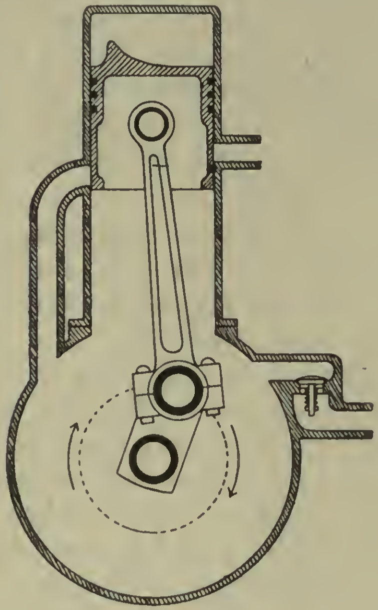


Fig. 3.

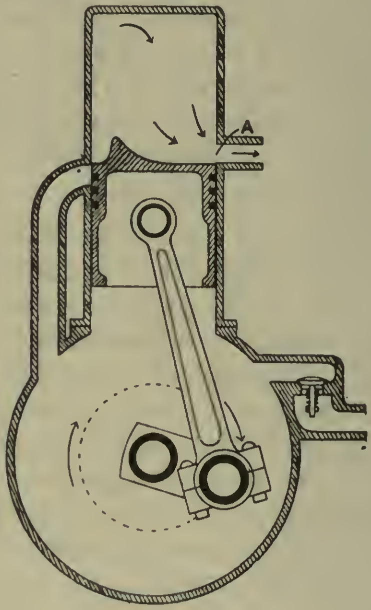


Fig. 4.

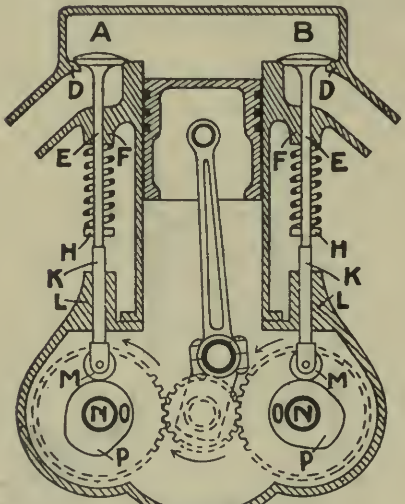


Fig. 5.

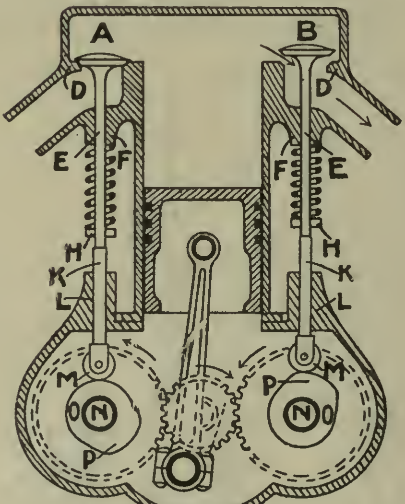


Fig. 6.

SCHEME OF OPERATION IN PETROL AND SEMI-DIESEL TYPE ENGINES.

this generally consists of an air compressor worked by hand, for filling the air tank for the first start. But the compression of air by hand is an extremely slow and laborious process and it should be avoided if there is any possible way of obtaining other power for the purpose.

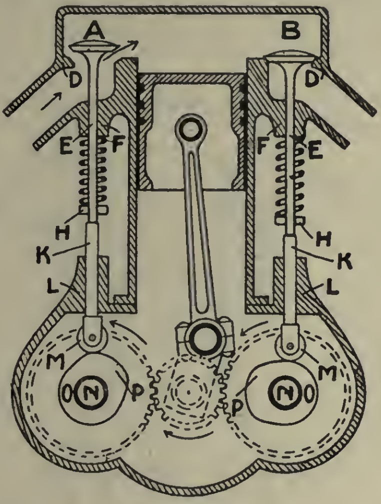


Fig. 7.

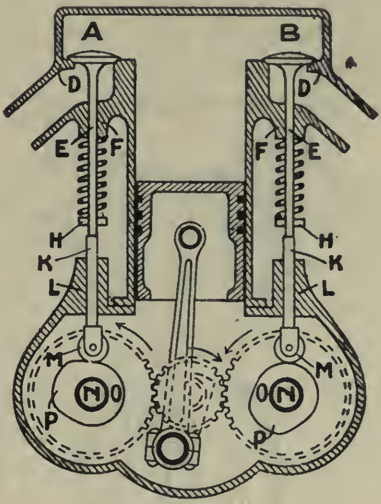


Fig. 8.

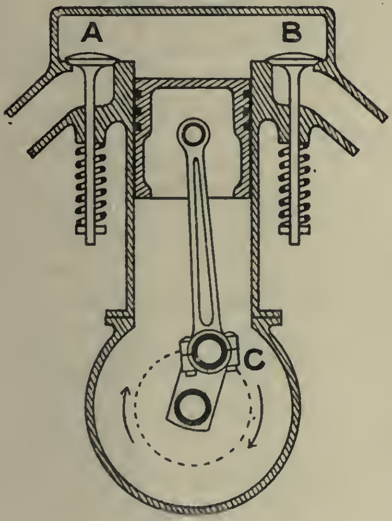


Fig. 9.

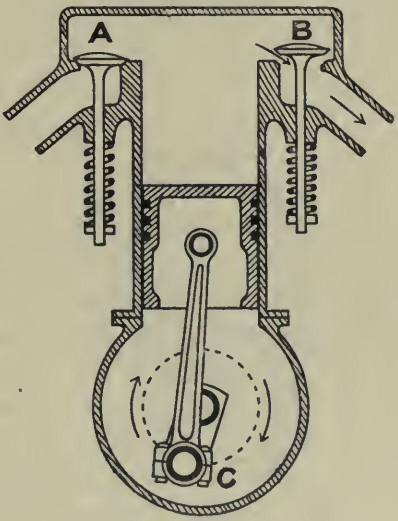


Fig. 10.

In an engine of two cylinders, starting air should be supplied to both, and in a four-cylinder machine to two at least. In those engines manœuvred by compressed air it is, of course, supplied to all cylinders, or should be. The air should be controlled by mechanically operated valves which will admit the air only through that portion of the piston stroke which is most effective, say, from about top centre through an arc of crank path of about ninety degrees. To go beyond this is unnecessary and also wasteful of air. The function of the compressed air or gas is simply to give sufficient momentum to the flywheel to carry it over until the engine fires, and the air should be shut off as soon as it fires.

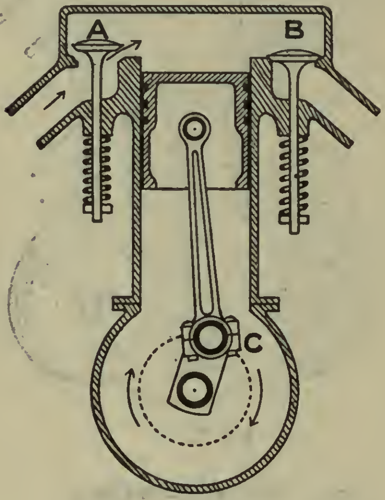


Fig. 11.

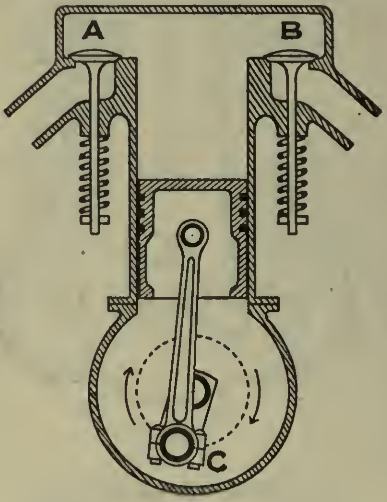


Fig. 12.

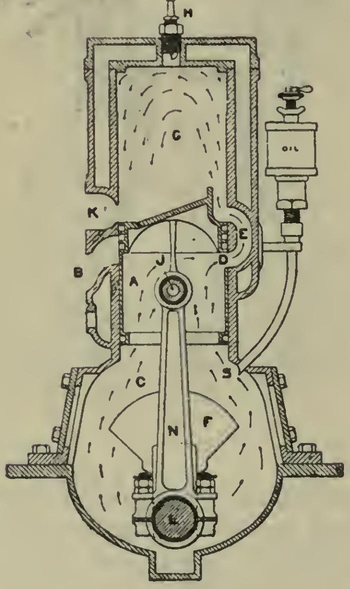


Fig. 13.

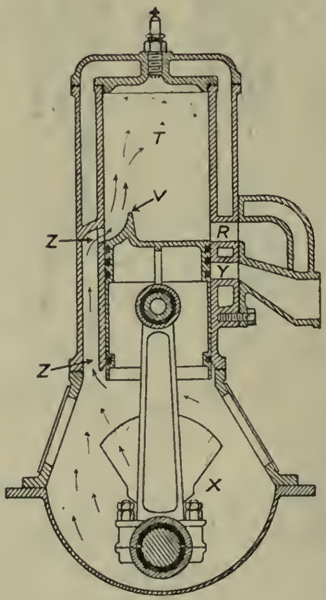


Fig. 14.

SCHEME OF OPERATION IN PETROL AND SEMI-DIESEL TYPE ENGINES.

EXPLANATION OF CUTS.

Diagrams one to four show a two-stroke cycle engine in section, in which Fig. I indicates the position of the parts when scavenging is taking place, with exhaust gas escaping on the right, while a fresh charge is entering the cylinder on the left, through the by-pass from the crank case.

Fig. 2 shows inlet and exhaust ports closed by the piston on its upward stroke, with inlet valve in crank case open to admit a mixture of air and gas, or of air only, into the crank case.

Fig. 3 shows the piston moving downward under pressure of the expansion stroke and, at the same time, compressing the ensuing cylinder charge in the crank case.

Fig. 4 shows the exhaust gas beginning to escape, before the piston has gone down far enough to open the inlet port on the left. The opening of the latter takes us back to Fig. 1.

Diagrams thirteen and fourteen show a three-port, two-stroke engine, in which Fig. 13 shows the position of the parts during scavenging. Here we have a short by-pass connecting the cylinder with the crank case through a port in the piston, while in Fig. 14 the form of by-pass is the same as that shown in Figs. 1 to 4. In both 13 and 14 the crank case inlet consists of a port close up under the exhaust port, as at Y in Fig. 14. This inlet port is kept covered by the piston, except that when the latter is near the top of its stroke it uncovers the port, thus admitting a charge to the crank case ready to be compressed by the downward movement of the piston.

Diagrams five to twelve illustrate the action of a four-stroke cycle engine. In Figs. 5 and 9 the piston is shown moving outward on its expansion, or power stroke, with both valves closed. In Figs. 6 and 10 the piston has started upward with the exhaust valve open to permit the escape of burnt gas, pushed out by the piston.

Figs. 7 and 11 show the piston moving downward on the third, or suction stroke, with the inlet valve open to admit a new charge to the cylinder. This valve now closes and the piston starts upward on the compression stroke, with both valves closed, as indicated in Figs. 8 and 12. This charge is then fired and the cycle repeated.

DISCUSSION.

The Chairman: I think the author of the paper has some little prejudice in favour of the two-stroke cycle, if so, it is up to some of the four-stroke advocates to make good their position. The question of packing is one that should appeal to many of our members, as it is impossible to be afloat long without learning something fresh about the vagaries of packing.

A Member: Referring to the following paragraph, where the author states:—" In gas, petrol and paraffin engines where the fuel is mixed with air before admission to the cylinder, throttling is accomplished simply by a reduction of the quantity of mixture taken in on the suction stroke, by a butterfly valve placed in the induction pipe. This valve controls the quantity of air admitted, and since the quantity of fuel entering the cylinder depends upon the air current, the richness of the mixture is not greatly affected by the action of the throttle valve."

Well, the carburetter jet is underneath this throttle valve and the flow of petrol from the carburetter depends on the creation of a vacuum in the induction pipe above this carburetter jet. Surely, if there is less vacuum in that pipe there will be much less petrol going into it, for as a rule or at least in most petrol engines there is an automatic air valve above the throttle, thus more air will go into induction pipe and there will be quite a variable alteration of the gas going into the cylinder. Throughout the lecture the author makes a lot of the dirt in oil and effect on valves, etc. Just below that the author says “If the air be uncooled, rapid compression of it to 150 pounds per square inch raises it to a temperature of 589° Fahrenheit.” I should like to ask the author if the fuel used is Diesel oil fuel, as it is called. Is that residual oil or gas oil?

The Author’s Deputy: Any oil; it does not matter.

The Member: The author refers to the utilisation of oil under boilers, and says this process is wasteful; but I should like to ask the author if it is possible to ignite, say, Mexican fuel residual oil at this temperature of 589° Fah. and use it in all Diesel engines. If it is not possible to ignite it and use it at that temperature then it will not be possible to use this residual oil in the Diesel engine. If you cannot use it in the Diesel engine the only thing is to use it under the steam boiler.

The Author’s Deputy: The vacuum in the carburetter if reduced by reducing the air current would take in less oil or less petrol. I think we can say it does. I do not know what variation in adjustment is practicable, but it does not seem to affect the mixture very much. Regarding compression—you can obtain in various engineering works a table showing the heat created by various compressions. A compression of 150 pounds, if the air is uncooled, will give 589° Fahrenheit. That will not ignite the oil, but it will vapourise it. In the Diesel engine you have a temperature of nearly 1,000° from compression alone, which is hot enough to ignite the oil. In the semi-Diesel engine you get your additional heat from the hot spot to ignite the charge. Compression does not ignite it, but the hot spot does. The hot bulb or plate is at a higher temperature. The usual term is to say it is “at black heat,” round about 900° Fahrenheit, and it is that which fires the charge.

A Member: I am surprised to learn from the author that the efficiency of an internal combustion engine is so very much superior to a steam engine; so very much so that it would appear that it is claimed that instead of being 100% it is up to 165%. I cannot see where that comes in. If we get a fairly good average for the steam boiler, giving 75% from a pound of coal by turning that into steam from which we get 65%. I do not think that 48% can be doubled by the internal combustion engine, over the present day steam engine practice for 100% efficiency; the author may explain that. The author surprised me when he upheld the two-stroke engine so highly, compared with the four-stroke. I am in agreement with him, it is the more flexible of the two. I should like to allude to what is mentioned about 4,000 pounds per square inch, and then as high as 8,000 pounds is mentioned. Is that an error, or did I misunderstand it? The reference is to pumping of fuel into an oval pipe. It seems to be an excessive pressure. Is there a fly-wheel on that engine? We steam engineers like an open tool. We like to see the moving parts and everything open. I do not see why the internal combustion engine cannot be open also. Is the two-stroke engine not capable of being built in that fashion? With all these arrangements of the cooling required it is a pity the engine has to go through such a process. From the great and excessive heat, from the firing of the fuel to the return of the scavenging it seems detrimental to the metal.

A Member: Can the author state if there is any definite advantage in hot plate over hot bulb ignition in the ordinary types of heavy oil engines, and if any wider range in engine speeds with either method.

The Chairman: A remark has been made upon the desirability of completely enclosing the internal combustion engine. This recalls a discussion before the electrical engineers. You will remember that the Willan's central valve engine, known as the C.V. engine, was totally enclosed, and a number of people made remarks that they liked to see the engine going round. The complete enclosure of the Willan's engine enabled the splash system of lubrication to be adopted, and so save labour as well as loss of oil. In the course of the discussion Col. Crompton replied to this criticism and asked the objectors if they had ever considered the case of the horse, which was a very complicated piece of mechanism, and if they thought he ought to be opened up to see his works in case anything went wrong.

I think one of the answers to the criticism is that if something does go wrong you have no time to avoid trouble, because of the high speeds of rotation at which these engines run. According to the power developed, speeds up to 400 or 500 revolutions per minute and sometimes as great as 600 revolutions per minute obtain.

In some of the smaller torpedo boats that I have served in, the speed of the steam engine was about 650 revolutions per minute, and the engineers used to feel the big ends in order to detect any symptoms of overheating, but it was done at the cost of saturated clothes. If the oil thrown about the engine-room in all directions had been caught by the casing enclosing the engine instead of the walls of the engine rooms and the engineer's clothes, and returned to the pins, there would have been more than enough to render the overheating of the pins impossible. It is this fear of overheating that leads many engineers to distrust anything that robs him of his power to feel the parts and so detect a trouble in time to prevent any untoward developments.

The AUTHOR'S DEPUTY: I do not think Mr. Bradbury meant to say anything against the steam engine; he was only making a comparison in regard to liquid fuel for generating steam when he said it took twice as much oil to get the same power from steam. So when you burn liquid fuels under boilers to raise steam you are wasting fuel. Regarding pump pressures, I believe that Messrs. Vickers from the start, with solid injection, have used about 4,000 pounds pressure with their pump. I think Dr. Hele-Shaw uses from 2,000 to 3,000 pounds in his little pumps. It sounds big on a small spindle, but it is not so much as it sounds. 8,000 pounds pressure refers to the Doxford engine, the best thing built to-day in the Diesel line, with one exception. This fires the gas at the ends of both pistons, but between the two. Their latest engine is of 2,700 horse-power on the crank-shaft, and the cylinder bore is only 22 inches or a little over, and the stroke is 3 feet 9 inches. At 77 revolutions it has a piston speed of only 576 feet, which is the lowest obtained in the type. There is only one other engine in the world to-day to equal it, and Doxford's have done the biggest thing in England in the internal combustion line. They only have 300 pounds compression pressure, and the fuel consumption is low; but they feed the fuel with force pumps, and these have a compression of anything from 7,000 to 9,000 pounds per square inch. One point is that the cylinder liners are only 1 inch in thickness, whereas in ordinary Diesel practice they would be something like $2\frac{1}{4}$ inches. The regulation rate of running is 77 revs. There would be no difference in the action of hot bulb or plate, if properly arranged. The point is, to water cool the hot spot sufficiently, and on the other hand you must not let it get too cool. If you keep the *point* that does the firing dull red it is all right. The better thing, I think, is a hot plug. But you want to arrange it not to get too hot, and you want enough metal to retain the heat when running light. You can run some of these engines a long time light by doing nothing beyond reducing the quantity of the fuel.

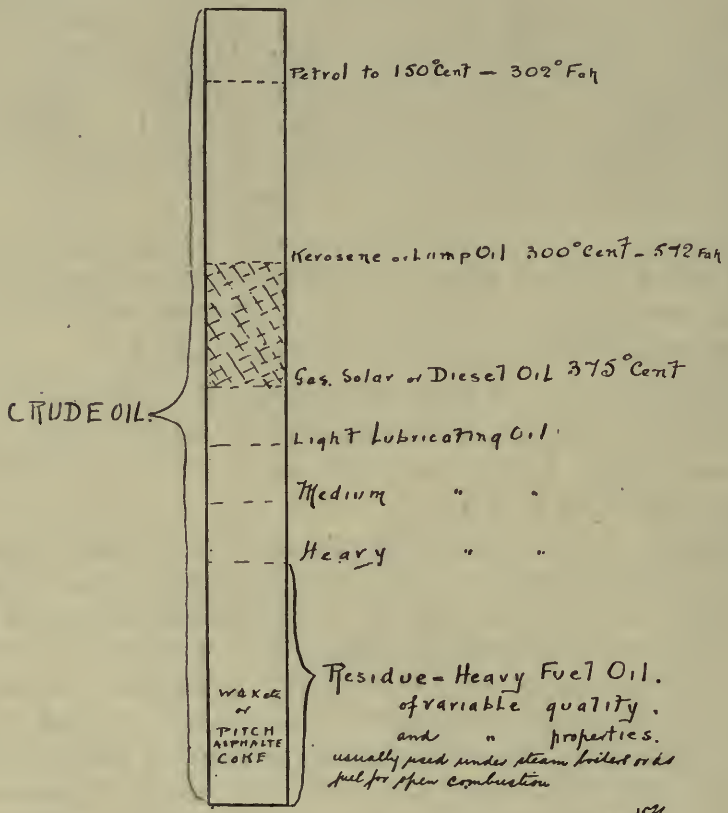
A MEMBER: In comparing the two and four-cycle engines, say of 1,000 I.H.P., I take it that the torque on the shaft would be much more uniform with the four than a two-cycle engine?

The AUTHOR'S DEPUTY: For the same I.H.P.

The MEMBER: Yes. Then the shaft and columns, etc., would have to be made stronger in proportion, which means that the four-cycle engine has some advantages over the two-cycle as regards weight per I.H.P.

A MEMBER: I was not quite satisfied with the answer given me. It might interest the author and also those interested, that to burn residual petroleum or the dirtiest material possible to get, without any elaborate cleaning of the oils, that Diesel engine oil is a distillate and distilled to an exacting specification, and is not either a crude oil or a residual oil. Also to learn that the burning of this residual fuel oil under the boilers of the *Aquitania* has reached such a fine point of efficiency that it is equal to exactly twice the weight of coal supplied. So, with these points of efficiency there is much more in burning this oil under boilers than people imagine. But the broad point is this: Is the author going to get a vaporisation point of some of our heaviest residual oils, containing pitch and coke? From all that we know of oil, certainly you cannot get this very heavy fuel oil with its high asphaltic composition vaporised and utilised in Diesel engines unless one anticipates a considerable amount of trouble from dirt and gumming up. I may make my remarks more clear by the aid of a blackboard diagram.

Further you have petrol which has to be distilled to 150° Centigrade; kerosene to 300° Centigrade; gas oil or Diesel fuel to 375° Centigrade. Then you have a heavy amount of valuable material making lubricating oils. Then we come to this, as fuel oil. What are we to do with it? We can use this Diesel oil in internal combustion engines and can get these lubricating oils which are very valuable and necessary. And then we still have this two to one equivalent to coal as fuel oil for consumption in the steam boilers.


Petrol to 150°Cent — 302°Fah
Kerosene or Lamp Oil 300°Cent - 572 Fah
Gas. Solar or Diesel Oil 375°Cent
CRUDE OIL
Light Lubricating Oil
Medium " "
Heavy " "
WAX etc or PITCH ASPHALTE COKE
Residue - Heavy Fuel Oil. of variable quality. and " properties.
usually used under steam boilers or as fuel for open combustion

Rough Division of a Crude Oil

INTERNAL COMBUSTION ENGINE FUEL cross-hatched

The Chairman: You are not thinking of thermal efficiency I suppose, of the engine as compared with steam?

The Member: My principal point of the efficiency of oil engines versus fuel oil under boilers is this: To get a satisfactory oil for internal combustion engines, this oil must be distilled; to distil it takes a certain amount of fuel to do so, and in the calculations of efficiency this cost of distillation (or loss of fuel to distil the oil) is neglected. From the commercial point, after the Diesel oils are distilled there are valuable lubricating oils and then the heavy residuals. Now the demand for oils is of such a nature that on the division of the various fractions we are left with a surplus of a material that would be impossible to get rid of unless we used it by open combustion such as under steam boilers.

The AUTHOR'S DEPUTY: The point Mr. Bradbury made was this—that the average pressure in the semi-Diesel engine is about 150 pounds isothermal, and 210 pounds adiabatic, and that will create a temperature of 589° in the cylinder, and he made the statement that this temperature would vaporise any oils used in these engines. In the Diesel, with a temperature of 1,000 degrees by compression, you should be able to vaporise anything. They do use oils so thick that they cannot pump them without warming. When you are burning the oil under steam boilers you have a larger opening in the nozzle, and you can blow dirt through that you could not blow through the internal combustion engine nozzle. Mr. Bradbury's point was this—it takes more oil under the boiler for a given power in the steam engine as compared with its use in the internal combustion engine.

The MEMBER: You cannot vaporise this heavy residue oil at 589° Fahrenheit.

The AUTHOR'S DEPUTY: No, you cannot do that.

A MEMBER: The diagram that has been put in chalk plainly shows that when all the distillates have been used in the internal combustion engine, then the old steam engine will come back again, because what is left at the bottom will be burnt under boilers. So there is a good thing in store for those who still stick to steam.

A MEMBER: I should like to ask the author how he proposes to have a pump without packing. It is with greatest interest I note from a good authority that the temperature of the circulating water can be maintained so high, viz., 180° F. to 200° F. and the engine work safely, also piston rings to give efficient results, the cooling water to pass exhaust manifold previous to cylinder jacket is commendable.

A MEMBER: There is nothing like paper for packing joints for oil—just ordinary paper cardboard.

The CHAIRMAN: It is easy to talk about cycles and the conditions to be fulfilled so that high efficiency may be obtained. But the engineer must, in addition, learn the exact settings and shapes of the cams, the best diameter and shape to give the valves, to resist the temperature stresses, the shape of the ports and passages, and so on for all the parts of the engine before a satisfactory engine is produced. Then the engineer-in-charge must know exactly how to detect any variations in working as they occur, and instinctively correct the valve setting or other cause.

The author refers to the question of scavenging, one that should have stirred up a question or two at least. In one place we are asked to contemplate the passage of the inlet and outlet gases, as orderly currents in the cylinder, but later on the risk of mixture that is disorder is admitted. At the speeds mentioned—1,600 feet per second—turbulence must occur.

It would appear that for perfect combustion the gas present at the instant of ignition will require its due proportion of oxygen, and that means, a certain weight of air must be drawn into the cylinder during the period that the inlet ports are open. The scavenging systems adopted by engine designers are intended to make certain that this weight of fresh air shall always be present, even when the two stroke cycle is adopted and the period of inlet and exhaust is seriously curtailed. As a matter of fact excess air is usually present, and the mixture in consequence of the turbulence of the gases due to the rapid motion of the piston will ensure quick combustion. A certain proportion of the products of combustion may be present at ignition and will not seriously impair the efficiency of combustion providing there is an excess of oxygen present over that needed for the perfect combustion of the charge.

Another statement in the paper that challenged criticism was the repeated use of adiabatic and isothermal pressures and adiabatic and isothermal temperatures in the cylinder at certain points in the cycle. You may express the pressure, temperature, or volume at any instant, in terms of different units, but their intrinsic values are the same. You know, of course, that an *isothermal* change of pressure and volume is effected at constant temperature where $PV = \text{constant}$, where $P = \text{pressure}$ at the same instant as $V =$ the volume, whilst an *adiabatic* change of condition is a process carried out, without the passage of heat across the boundaries of the working fluid during the process, and in this case $PV^{\Upsilon} = \text{constant}$. Both of these processes are impracticable, and the actual changes in the internal combustion engine must be intermediate so that $PV^{n} = \text{constant}$ represents the actual change where n it less than Υ and greater than 1.

When an engine runs very fast then the value of n approaches, the value Υ, because the loss of heat during the combustion and expansion stages by conduction and radiation are reduced, but if the engines run slowly then the greater time occupied by the expansion stage, permits of the heat losses to be much increased.

For these reasons in computing pressures, volumes and temperatures it is surely better to give to n a suitable value, rather than estimate the extreme limits possible. In gas engines working on the four cycle the value of n is round about 1·3.

Another point which frequently arises in these discussions is the value of oil as a fuel in place of coal. The efficiency of the boiler varies with the efficiency of combustion, and in the case of oil fuel this is easily maintained at a high figure with quite ordinary care, hence probably a safe figure to take for our present purpose would be round about 80 per cent., it certainly could be very little greater than this. Turning to the engine using 11·2 lb. of steam per I-h.p. with an efficiency of 19·1 per cent.—then the total efficiency over all would be 15·3 per cent. Now using the same oil in a Diesel engine a thermal efficiency of 40 per cent. and upwards may be expected, showing an overwhelming gain by utilising the oil in a suitable oil-engine, over burning it in the furnace of a steam boiler.

Carnot long since showed that the limits of efficiency in heat engines depend upon the temperature range available, but it is difficult to realise this fact fully.

Assume the temperature in the internal combustion at ignition to be 2800° F., and that of the charge to be 200° F., then the utmost fall of temperature possible is $2800 - 200 = 2600°$—hence the theoretical efficiency would be $\frac{2600}{3260} = 0·794$ or 79·4 per cent., of which actually only about 36 per cent., would be realised. In the case of a steam engine using steam at 180 lb. pressure per sq. in. and superheated say 100° F. and exhausting into a condenser at a temperature of 120° F. the steam temperature would be 473° F. and the drop of temperature is then $= 353°$, so that the greatest efficiency conceivable would be $= \frac{353}{933} = 0·38$ or 38 per cent., of which about 19 to 20 per cent. would be realised. These figures should convey some idea of the reason for the enormous advantage of internal over external combustion Heat Engines.

The AUTHOR'S DEPUTY: Referring to the remarks on packing. There is one well-known oil engine which uses leather, and it works very well, until the leather perishes. Asbestos is as good as anything you can use for packing. I can recall two pumps working on oil engines that have not been packed since before the War; and we were shown an unpacked pump from a Bolinder engine, that had been working for five years without leaking. Some engineers think the only thing for the body of the pump is cast iron, and others use gun-metal. Whether they use one or the other the barrels should be lapped perfectly smooth, and the plungers should be case-hardened and ground true, and should fit with a tight working fit. In the Doxford engine they say the pump plungers are a "hard push fit." So that gives an idea of what packing is required in this sort of work. As to adiabatic and isothermal terms, I think they are only comparative. They are only theoretic. If you could compress air to a certain pressure without creating any heat in compressing it, you would say you had isothermal compression of so many pounds per sq. in. But every time you compress air you create heat, and whatever heat is added to the isothermal temperature would be termed adiabatic heat, I imagine, and the pressure per sq. in. would be termed adiabatic pressure. There are certain rules for estimating what that heat would be, if uninfluenced by any extraneous feature. I think the adiabatic pressure is perhaps 41 per cent. added to the isothermal. Thus, if you had a compression pressure of 200 pounds isothermal, it would actually show on the indicator about 282 pounds per sq. in., while the temperature would rise to 672° F. If you squeeze the air up slowly you would not get that much heat. Regarding the field covered by this paper: It was based on the Syllabus (Section 4). of which, unfortunately, we have no copy to-night, and the author endeavoured to cover those points. That is why it covered rather a wide field. I think perhaps it would have been better to have started with something like this, for those who wanted to study the subject. It is impossible to get a paper up that will give everybody a considerable amount of information. We felt that if we could get two or three new points fixed in the minds of those listening, we should have gained something. But the chief thing is to get people to think for themselves.

A member calls attention to the following:—

In the discussion on the Paper "Fuel, Valves and Ignition" reported in the March Transactions, I notice on page 412 a question by a member in which it is assumed that the torque on the shaft of a four-cycle engine is more uniform than in the case of a two-cycle, and also that the Columns Shaft, etc., would be lighter for a four-cycle engine than corresponding parts for a two-cycle.

As these assumptions were not corrected in the report, some members who are not familiar with the two types may get some confused ideas. As a two-cycle engine has a working stroke in every two strokes, whereas a four-cycle engine has only one in every four, it is obvious that the turning effort in the case of the two-stroke must be more uniform than in the case of the four. A comparison is made in a booklet published by the "Sulzer Engine Co." between two 1,100 B.H.P. engines of the two types, and as this is nearly the size mentioned by the member, the figures given may be of interest. Diameter of shaft according to Lloyds' requirements: Four-cycle 15 in., two-cycle 13½ in. Total weight of installation: Four-cycle about 530 tons; two-cycle 320 tons.

Mr. A. W. **Bradbury**: To put the matter perfectly plain and replying to Mr. Kingdom's question, I would say that members who are not familiar with the two types of internal combustion engines may be somewhat confused in a statement regarding the torque on the shaft and the weight of the reciprocating parts; and as a two-cycle engine has a working stroke in every two strokes, whereas the four-cycle engine has only one in every four, it should have been stated that the turning effort in the case of the two-stroke would be more uniform than in the case of the four, and that the connecting rods, shafts, etc., would also be lighter for the two-stroke type. A comparison is made in the Sulzer Engine Company's booklet, and I cannot do better than repeat what Mr. Kingdom has said, that the comparison given between two 1,100 B.H.P. engines of the two types are of exceeding interest. They give a diameter of shaft according to Lloyds' requirements, four-cycle 15 in., two-cycle 13½ in.; total weight of installation: four-cycle about 530 tons and two-cycle 320 tons. I think that Sulzer comparisons may be taken as being correct, as they are absolutely reliable in data given of their engines, and know as much about two-stroke engines as any maker in the world.

SESSION 1919-20.

President: LORD WEIR OF EASTWOOD.

The Lubrication of Diesel and Semi-Diesel Engines.

BY MR. EDMUND G. WARNE.

Read Tuesday, December 14, 1920.

As a preface, before starting this lecture I should like to refer to one point, which is, that I feel it is quite impossible for me to deal exhaustively with the subject of the lubrication of Diesel and semi-Diesel engines in the course of a lecture. Much depends on the engineer; there are no two systems of lubrication alike on any two makes of engine, and I think it is all to do with the way the engineer in charge sets about matters as to whether the machinery is going to run rightly. From a practical standpoint it means nothing more or less than investigating the system as it stands on the engine, and going through it from end to end. If that is done it pays very greatly, as I have found to be the case.

Lubrication problems have played a part of no mean importance in connection with the successful development of both Diesel and semi-Diesel (or hot-bulb) machinery. Perhaps with regard to the former type of plant the solution has been comparatively easy, on account of the straightforward method in which it is possible to deal with the question of an efficient supply of oil to bearings; however, as concerns the semi-Diesel engine, a set of conditions may be encountered which demands the application of special measures.

No matter what the particular means of lubrication employed, there is not much doubt that the engineer who, in the course of practical running of machinery, makes it his business thoroughly to investigate the lubricating arrangements fitted to the plant under his care is unlikely to experience trouble. Difficulties are more likely to arise when the guiding light of the engine room—as soon as the telegraph has settled down to full or half ahead, as the case may be—assumes an attitude of indifference and leaves matters unreservedly in the hands of Providence. The actual cycle of operations in an engine depends more on the machinery than on the engineer, whereas the lubrication of bearings (although in many cases automatic, or partly so) calls for a certain amount of personal attention, and I need hardly remark that it will pay for ungrudging care many times over.

These considerations of investigation and attention apply especially to the average two-cycle crankcase-compression semi-Diesel engine. It is not one of the best features of this type of plant that in most instances an ample supply of lubricating oil under pressure is practically out of the question. We find the oil delivery almost cut down to a minimum, whereas the natural desire of anyone who has experienced the invaluable benefits of forced lubrication is to increase the supply to a maximum. Regrettably enough, this is difficult to achieve, on account of this system of crankcase compression—a system which, notwithstanding, is widely used. The air carries practically all the " free " oil in the crank chamber to the combustion space, where the inevitable happens. In fact, if oil is present in anything like an excessive amount the engine will " run away " on it, possibly with unlooked-for results. This is, of course, an exceptional occurrence, and it may be mentioned that the most effective means of stopping the machinery is by holding open the crankcase air valves, if possible.

In the case of the usual type of semi-Diesel engine, then, a compromise has to be effected. Generous oiling is to be avoided, whilst restriction of supply beyond a reasonable limit is followed by serious wear of the bearings, if not actual breakdown. As a means of relieving the situation the plan of isolating main bearings from the crank chamber is one to be commended, suitable baffles being provided for the two-fold purpose of preventing escape of air from the crank chamber during the discharge stroke, while, in addition, avoiding ingress of lubricating oil from the main bearings, along the short length of shaft between.

It remains to be seen whether designers will rest content with the present system, carrying out such improvements and modifications as suggest themselves from time to time, or whether the crankcase-compression engine will, ultimately, give way to a design on the lines of a two-cycle Diesel type of plant fitted with scavenging pumps. As regards most of the present forms, no doubt the comparatively low cost of construction will assist towards prolonging the life of this machinery for many years, but I cannot help thinking that the crankcase-compression engine is nothing more nor less than an expedient. Nevertheless, it is to be treated with the utmost respect, and the present-day engineer will not wish to sever connections with it on account of what may happen in the future.

At least one type of semi-Diesel engine is to be found which has, I believe, already proved its worth, and is arranged on the open crank-pit system. It conforms to steam practice in some respects, inasmuch as crossheads and guides are fitted, while important bearings may be seen and (what is more important) felt. In the particular case under review a drip-feed system of lubrication is fitted, and the qauntity of oil supplied to main bearings, crankheads and crossheads needs no special consideration beyond such attention as would be given to the average steam engine of equivalent horse-power. From the lubrication point of view this arrangement is entitled to a measure of commendation, although I would like to see the system carried out to a more advanced degree, a direct pressure-fed oil supply introduced, and closing in the lower part of the machinery to an extent which would prevent splashing of oil, while at the same time draining off the surplus to a working tank in the circuit, as is usual with most present-day Diesel machinery.

In the foregoing remarks I have given some indication of the broad lines on which current practice is instituted, and at the same time endeavoured to hint at future designs, but it will be recognised that a great deal depends on circumstances. Cost of construction, while not everything, is a very serious item, and one which is not likely to hasten the advent of the forced-lubrication two-stroke semi-Diesel engine. At the present market prices of machinery, uncertain developments, strikes, rumours of strikes and other disturbing factors affecting trade in general, we may fully expect to have to deal for a long time to come with the semi-Diesel engine as we find it now, and not as we would wish to have it. In the meantime builders, we imagine, are not entirely unaware of the possibility of improvement, and we must rest content to concern ourselves at this juncture with lubrication systems as applied to modern, and perhaps somewhat restricted, developments.

THE LUBRICATION OF DIESEL AND SEMI-DIESEL ENGINES.

A good many practical suggestions have been made of late concerning the general running of semi-Diesel machinery, and it seems to me to be fair to assume that such definite propositions as have been advanced are sufficiently disinterested to warrant investigation. On points of general running, most engineers are agreed that modern plant of this type is of an excellent character. These seems to be little difference of opinion as to lubrication: it is not difficult to detect condemnation of a good many features of modern practice, but at the same time the suggestions for improvement do not always come along at the same rate. I think that engineers and drivers of semi-Diesel machinery should not hesitate to propose any improvements in existing practice that may appear beneficial to users generally.

As far as the average type of marine Diesel engine is concerned, the lubrication system settled down to a basic principle of arrangement in the early days of its career, and in most cases the system employed follows forced-lubrication steam practice to a great extent. It appears to be fairly well established that the most workable scheme comprises a main pressure supply from which is tapped a series of connections leading to each main bearing. Around the circumference is turned a groove in the metal coinciding with the point of entry of the oil. At any position on a coincident line around the circumference of the shaft a hole is drilled to the centre, meeting the central hollow portion of the shaft itself. The crank web is drilled (and, of course, suitably plugged afterwards) while communication is established between the hollow crank pin and the interior of the main shaft. The same process is followed out in the case of the crankpin and crankhead bearing. From the latter the oil has to be conveyed to the gudgeon, and in order to effect this the connecting rod may be drilled through its length (if not originally hollowed) or alternatively an external pipe may be fitted.

Thus a state of pressure exists from main bearings to gudgeons, while if the engine is of the crosshead type the oil is conveyed through to the guide surface, thence draining down to the crankpit. This pressure is chiefly dependent on three items. First of these is the pressure in the main supply from the pump. Next in order of importance is the degree of adjustment of bearings, whilst a third factor present in the course of ordinary running is the temperature of oil in the system. From a practical standpoint I am leaving out considerations of quality, viscosity, etc., of the oil used and assuming that a reasonably suitable lubricant is employed.

On the question of pressure it seems to be a kind of unwritten law that 10 lbs. per sq. in. on the gauge is—if not precisely a red line of demarcation—at least a happy medium, and I do not know that this more or less accepted standard needs or deserves any adverse comment. It has stood the test of time and is therefore a figure to be treated with due reverence, but at the same time I believe that on account of the nature of its origin, *i.e.*, in connection with high-speed steam-driven dynamo or similar practice, it may become customary to lower the figure slightly, regarding 10 lbs. as a maximum. As a reason, I would point to the tendency to lower the rate of revolutions of marine Diesel plant, while at the same time bearing in mind that an excess of pressure is not to be aimed at. The considerations are mainly with reference to the maintenance of a film of oil between journals and bearings under load.

Reversal of pressure, to some extent a deciding factor in lubricating problems, has a bearing on the matter, and some doubt has been cast on the high-powered two-stroke Diesel as regards capability of standing up to its work. It is rather early yet to express an opinion, and in any case mere prophecy is of little value, but I believe indications go to prove that, contrary to some previous expectations, a high bearing oil pressure is not necessarily demanded, provided that the bearings are properly constructed; possibly, more than one firm constructing two-stroke marine Diesel plant up to maximum present-day powers has already decided that abnormal oil pressures are entirely unnecessary. The design of bearings is a much more scientific performance than was imagined to be the case some years ago, not only with regard to dimensions, but also with respect to the admission and retention of the oil film.

A good many marine engineers are by this time acquainted with the Michell thrust block, and I know of no better instance of efficient design. It was established by Prof. Reynolds that the two surfaces of a journal and bearing must incline towards each other in the direction of motion, so that the oil film could take up a tapered formation, and this principle is followed out by the Michell thrust and ordinary journal bearings. Time does not permit of an investigation into the special considerations involved, but I mention this particular construction because it shows what can be done by careful design (and, incidentally, without forced lubrication). I must leave this interesting subject here, but at the same time I should recommend every marine engineer to take an opportunity of investigating the design, because it carries with it one or two useful lessons in lubrication, quite apart from the particular form of construction employed.

Returning to the subject of the usual type of bearing and its treatment; provided the surface has not been disfigured with a wonderful network of unnecessary oil grooves, it will stand up to its work in a satisfactory manner, given enough oil. The advent of forced lubrication assisted, in a large measure, to prove that a bearing, of which the oilways when placed end to end would reach from the earth to the moon—more or less—is not the last word in efficiency. On the contrary, with a pressure supply on the lines indicated no oil grooves are required, beyond those necessary to convey the lubricant from bearing to bearing, and at the same time the clearances can be, and should be reduced to a minimum. One or two moderately slack brasses will have the effect of lowering the oil pressure to an appreciable degree as soon as the circulation has proceeded for a short time and the temperature has risen a few degrees, whereas an engine in which the bearings are up to the safe working limit, in the first place, will keep a satisfactory pressure for months on end.

As far as the pumps themselves are concerned, the design is of small moment. The general practice appears to be that engine-driven plunger pumps are fitted to smaller plant, whilst the high-powered modern marine Diesel engine takes its oil supply under pressure from ordinary gear-wheel pumps, usually in duplicate, or, in some twin-screw vessels, triplicate. They are mostly electrically driven and work excellently. Plunger pumps may of course be driven from any convenient portion of shafting, having due regard to speed and capacity of the pump. One system is perhaps worthy of comment, in which the pump is directly-driven from an eccentric on the main shaft, whilst the plunger is bored up its length and discharges firstly to the eccentric strap, thence through a hole in the sheave to the main shaft, bearings, crankheads and gudgeons—not a single pipe being fitted to the system. However, this arrangement is, no doubt, somewhat uncommon.

That the working temperature of the oil has a decided bearing on the efficiency of the engine has been realised for some considerable time, and there are definite indications that considerable importance was attached to this feature by German engineers in their development of the submarine Diesel engine, during the war. As far as our own activities in research work are concerned, a good deal of private and Government experimentation has been carried out from time to time. Some recent tests at the National Physical Laboratory in connection with a worm gear testing machine show that with a pure mineral oil an efficiency of 95·6 per cent. was obtained, whilst at 52 degrees Centigrade the efficiency fell off slightly, dropping to 93·5 per cent. at 75 degrees Centigrade. Between 50 degrees and 62 degrees Centigrade, violent oscillation of the gear rendered readings difficult to obtain, and it is interesting to note that the critical temperature remained the same when the load was doubled.

This critical temperature is about 71 degrees centigrade for cylinder oil, and a characteristic of all mineral oils appears to be that the efficiency drop occurs suddenly at a particular heat. Sperm oil shows a gradual decrease in efficiency as the temperature rises, while castor oil, as might perhaps be expected, gives the best results, rape oils being next in order of efficiency. The value of adding graphite to oil has also been the subject of investigation, and it appears that while animal and vegetable oils show but faint signs of improvement, with mineral oils the addition of graphite has, in some instances, a beneficial effect. In the case of a hot bearing, graphite has often proved useful, but its employment must be attended with care, and very moderate quantities only should be employed on account of the risk of choking oilways. As regards cylinders, I have not heard of any experiments with Diesel and semi-Diesel plant, but it seems not improbable that graphite might usefully be employed in small amounts. Concerning petrol and paraffin machinery, its use has been attended with excellent results, as it is unaffected by heat, and not only lubricates the piston rings, but assists towards filling up minute blow-holes, scores, etc., and thereby increasing the compression in a badly worn engine.

In the ordinary way, cylinder liners of both Diesel and semi-Diesel plant are generally lubricated through an encircling ring of piping provided with branches which communicate with holes in the cylinder. These branches, of course, run through the jacketing space; the oils used are of various qualities, but with a new engine a low viscosity oil is most suitable on account of the comparatively small clearances, while as these increase a more viscous oil may be employed. A comparatively high flash-point is desirable, in order to avoid the risk of explosion in the crankcase should the gas blow past the pistons and rings; but of course this qualification does not arise in the case of large engines fitted with crossheads, guides, and isolated cylinders.

Air compressors demand care in the selection of oil, as the corrosion of intercooler coils has been traced to organic acids formed by oxidation of the lubricant. Generally speaking, a mineral oil of low specific gravity and high flash-point is considered suitable for the purpose. Testing the flash-point and viscosity is rather out of the scope of the practical engineer on board, but I may perhaps mention that as far as viscosity is concerned it is possible to test this by an instrument of simple construction and approaching the accuracy of laboratory devices. However, as the viscometer I have in mind is somewhat expensive, and, moreover, testing of this description is out of the real province of the driver, further reference need not be made to it here.

Lubricating oil pipes and fittings are fruitful sources of trouble unless fixed with care. Throughout the engine no system of piping better exemplifies the proverb concerning a stitch in time, and when fitting up after inspection or repair it should be a hard and fast rule to blow through all pipes, union connections, etc., or, alternatively, pass a wire down, in order to ensure that there is complete freedom from foreign matter. When dismantling, if the pipes are to remain off the engine for any length of time, it is a paying proposition to bind up the male screwed ends with twine. Nothing is more annoying than to find a damaged thread, especially if the nut has to be screwed up in an awkward position, and in fact many a thread has been stripped through a burred edge giving the nut a false start. If short union connections are fitted on main or other bearings they should be tightened well home—preferably making a metal-to-metal joint; if this is not done, as soon as the union is slacked for dismantling in all probability the nipple will twist off unless observed in time.

The practice of soft-soldering lubricating pipe connections is one to be avoided, excepting in an emergency. Brazing is the most suitable method and assists in prolonging the life of the connection. Vibration will quickly find the weak point of a pipe, and if the joint is out of sight serious trouble is bound to develop should the pipe part.

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Centrifugal lubricators and their connecting pipes for crank-head bearings demand special attention if the risk of running the metal is to be avoided. Unless spigoted in such a manner that incorrect fitting is practically impossible, the banjo should be tested circumferentially and laterally by fixing a wire on the adjacent bearing or other convenient position and turning the engine a complete revolution with the turning gear, if there is any doubt about the fitting being perfectly central. In any case, after the banjo is in place and the lubricating pipe is in its correct position, the engine should be turned and the arrangement checked. At least one type of engine is designed with a lubricating system of a different nature, *i.e.*, the crank pin receives its oil supply via the gudgeon, through a hollow connecting rod, thus avoiding the use of a banjo lubricator.*

Oil pipes to centrifugal lubricators may be made of steel, with advantage, and a more or less general use of steel piping is often advocated. The disadvantages are obvious, and include less flexibility and greater liability to rust.

Probably the most common form of lubricating the gudgeon in a semi-Diesel engine is through a pressure fed pipe leading through the liner and coinciding with a hole in the pin itself. Certain designs do not provide for adjustability of the gudgeon brasses, so that wear can only be remedied by fitting a new bush; in any case, this wear can be minimised by paying due attention to the oiling, and as a matter of fact a gudgeon will readily seize up if the engineer allows the oil feed to fail.

The oil pumps supplying the various bearings are usually efficient in working, but nevertheless pay for attention. The discharge is generally variable, and adjustment may be effected by altering the length of the plunger-stroke. In practically every case this may be performed while the engine is running, and the number of drips per minute checked through a sight-feed. There are many different makes of pump, but I do not think that the majority present any difficulties to the least experienced driver, provided he gives them a careful preliminary survey. A great point is cleanliness in the well of the pump, and oil should be strained before filling; moreover, a strainer should be conveniently fixed on the filling orifice, or across the top of the box if the whole cover is removable. A periodical clean out is advisable, but on no account should the box be wiped out with waste. Swilling out with paraffin is a good plan, after which the box should be left to drain.

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*Since the foregoing was written, another system has been introduced for semi-Diesel plant. The air pressure in the crankcase forces oil from a container to the main bearings, thence to the crankhead and gudgeon, and except for a small matter of detail, the system corresponds with forced lubrication Diesel practice.

Operating mechanisms for pumps are often provided with adjustable ratchet-feeds from the shaft, and many cases of failure have been traced through carelessness in watching the drive, which, if the teeth or pawls become worn, may cease. Generally the pawl wears, and it may be mentioned that an engineer who recently had to deal with eleven cases of crankhead failure found that seven instances were due to this cause. The average semi-Diesel engine is not prone to breakdown, but it has been stated in more than one quarter that ninety per cent. of failures are due to lack of attention and failure of the oil-drip. The engineer who allows himself to become lax in the matter of attention to lubricators gets himself a bad name, but what is considerably worse, he may also get the engine a bad name, and I am afraid that it is only too true when it is stated that the more careless the engineer in regard to lubrication the more thorough is his abuse of the engine when a breakdown occurs.

It is recommended that ratchet wheels of lubricators be painted, say, with a line of bright red across. They can thus be more readily observed, and stoppage is likely to be noticed almost immediately.

Water and bilge pump eccentric straps often come in for more than their share of neglect, especially when grease lubricators are fitted. A grease lubricator has a comfortable look about it; one hopes it contains grease, but the lubricating properties of hope are not pronounced. I do not think I am alone in advocating that a worsted-feed oiler on pump straps is a better fitting; the oil cup and needle valve arrangement is also fairly reliable, but it has the disadvantage of being affected by vibration, and may even close off entirely if insecurely locked.

In conclusion, I would like to lay stress on cleanliness. Dirt is the doom of efficient running. The crank chamber must be clean, and above all free from particles of waste, while the same remarks apply to oil boxes. The presence of water is naturally to be avoided, perhaps specially with regard to worsted-feed containers, where its presence is often unsuspected for considerable periods.

The Chairman: In my own experience I have seen great savings effected by mechanical lubrication from hand oiling in the case of small internal combustion engines, in some cases saving half-a-gallon an hour. The limitations of the closed crankcase engine are also commented upon. It is interesting to find that the author has found that engines with forced lubrication have not made an unusual amount of splash or dirt when not covered in.

A Member: I do not think it is correct to impute praise to the Germans for the idea of using thermometers in connection with lubricating systems, as I think for a long time past it has always been the practice in the Navy very extensively to use thermometers in all parts of the engine. No doubt in some cases they have been supplied to ships where provision has not been made to use them to the best advantage.

The Author: I had in my mind a comparison between British and German submarine Diesel engines, and I found that the system was carried out more thoroughly in the case of the German submarine Diesel engine.

A Member: Everything seems to depend upon getting the proper oil, and I quite agree with the author of the paper, read here November 16th, that the viscosity of the oils should be known in order to determine whether the fuel can be handled without heating or not. I therefore suggest that the engineers in charge of all oil engines ought to be able to test for viscosity and flash-point, etc.

This is a very simple thing to do, and would save a lot of trouble, and I should have preferred the writer to have emphasised this point more fully, because all the care in the world, on the part of the best possible engineer, is so much labour lost if the oil in use is not up to the standard. I cannot say that forced lubrication is quite perfect yet.

I had one instance during a critical time in the late war when an auxiliary engine stopped dead and couldn't be started. After taking the doors off it was found that the metal had run out of the bottom end. This was the effect, but the cause was very difficult to find out; the pressure gauge showed 10lbs. up to time of stoppage. The crank shaft had to be taken out, and the white metal drilled out in the oil holes. I may say the oil was of good quality and well filtered, and was running all right in another closed engine beside it. The only conclusion that I came to was that some piece of white metal had come off and blocked up the oil-hole. The author speaks of thermometers, etc.; I am in favour of these but it would be difficult to fit one to the bottom end of a connecting rod in an enclosed engine; it would not give the engineer on watch any notice, simply because he couldn't see it.

This particular engine I have just mentioned had the oil pump in the corner of the crankcase (the most awkward corner to be correct) and to overhaul it was a very difficult job. I should like to suggest that oil pumps in this class of engine should be fitted outside and worked with some kind of gearing so that the speed of the pump could be increased in the hot weather and reduced in the cold weather. The suction pipe could always be felt by the engineer on watch, and this would be an indication of how the bearings were working. Thermometers could be fitted here as required.

The discharge side of the pump could be fitted to lead to the different bearings, and a sight feed could be fitted showing the amount of oil going to each bearing, with taps to regulate the flow of oil as required.

The CHAIRMAN: In regard to water drip from which we get water interfering with the lubrication in the oil engine:—The trouble is common in steam, but in an oil engine it might be useful to mention where we should look for water interfering with the lubrication.

The AUTHOR: It is chiefly in connection with outside bearings, and I am speaking more particularly of the small shafts, as well as the main shafts, to which no pressure supply is made, and ordinary worsted-feed boxes are fitted; possibly there is a drip from the circulation system; then you get the water at the bottom of the boxes, and do not know it is there. I have known cases where this has gone on for some time. The bearings have been in a bad state, yet the boxes always showed a good quantity of oil.

A MEMBER: I think we can understand that in the working of the engine there should be no water in the ordinary way with enclosed engines. But if you take the case of a small engine-room when the vessel is at sea and jumping about, you will get water into the bilge and that will find a way into the oil. I do not know any precaution you can take that will prevent water having access to the oil under varied sea conditions.

The AUTHOR: I think it is a good system to go round with a syringe and dip the syringe to the bottom of the boxes, thus using it as a pump to test them.

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There are many things I should like to have expounded in order to have made the subject more complete. On the question of forced feed lubrication I must say that I had most decidedly advocated forced feed lubrication for Diesel engines, but my point was that one could hardly apply it to semi-Diesel engines. As regards the pure Diesel engine, I do not think there is anything better. My experience in connection with the operation of Diesel engines is that having a respectable forced feed lubrication system your time may readily be occupied with other things. Once you have got rid of the idea of hot bearings then you can consider the points of operation of the engine which should be attended to. The man who is always rushing round with an oil feeder and a bucket of water cannot pay so much attention to the general running of the plant. In regard to the elimination of oil grooves; with a forced lubrication system oil grooves are unnecessary. I think that has been very conclusively proved. The oil gets out of the bearings fast enough without any oil grooves to help, and it will find its way into the bearings in a most satisfactory manner. I have seen gudgeon pins knocked into white metal bushes with a 14 lbs. sledge hammer and the engine has run perfectly without any oil grooves at all. That has led me to believe that oil grooves are rather a mistake. On the question of working temperature, I do not think I can add anything to the remarks made. It is a matter of great importance, and why I mention that the use of castor oil might be expected to give good results was because it usually does give the most satisfactory results. If you can get castor oil to use with a hot bearing, and you use that oil instead of the ordinary mineral oil, you stand a chance of cooling the bearing down very much more quickly.

A Member: The author made slight reference to straining the oil before filling the oil boxes, but we would like to have heard much more of this extremely important subject. Obtaining the very best quality of lubricating oil will not itself avoid trouble; it should be passed through a filter and refiltered at least two or three times before getting to the part lubricated. Most lubricating boxes and receptacles have gauze wire over the divisions, which are very inefficient, and in most cases will allow fine grit to pass. Again, one will often see the can containing the oil for filling up boxes, etc., standing on a bench or seat entirely open to the atmosphere and sometimes close to the dirt and dust from the engine-room ladder. If these were prohibited in an engine-room and only cabinets with tops so arranged that they could not be left open, were used, many of the lubricating troubles would be avoided.

The filling cans should have a fine filter medium fitted in the spout, so arranged that it is easily removable for cleaning purposes. The filling cans themselves should be left upside down under cover of the cabinets.

No doubt the many readers of this paper would be pleased to hear more on the quantity of oil to be used in different parts of a well-designed engine. Perhaps the author will take, say, six different kinds of engines and five or six different parts to form such a table. One of the greatest difficulties a marine engineer or driver has in working an oil engine at sea is to know how much or how little oil should be used in the various parts.

The author has told us of the disadvantage of over-lubrication, and we all know the disadvantage of under-lubrication. No doubt other contributors to the discussion will give us of their practical experience, especially with regard to the lubricating oil that should be used, and whether single service oil should be used for top and bottom ends or whether a type of cylinder oil for the former, and engine oil for the latter has been found more satisfactory.

The Author: This member has emphasized the necessity for straining oil, and has indicated that he would like to have heard more on the subject. To bring the whole, or perhaps even part, of the many questions affecting the subject of oil engine lubrication within the scope of a lecture means brief reference to many interesting points, and I have found it necessary to limit myself in many ways, in order to deal as generally as possible with what appear to me to be the more salient features in connection with the subject.

Regarding the suggestion that a table might be drawn up, taking six different kinds of engines and various parts of the mechanism, giving the quantity of oil to be used at each point, I do not think this would be quite practicable. No two engines use the same amount of oil (at least, that is my own experience), and any attempt to tabulate quantities is likely to have bad results in some quarters. One cannot safely tie down a driver too closely to a fine measure of oil consumption. It annoys the driver, never has paid, and never will pay. On the whole, I think such a table as suggested would serve little practical purpose. Needless to say, I am looking on the question quite from the point of view of running the engine under ordinary seagoing conditions.

A type of cylinder oil for top ends would no doubt prove more satisfactory than engine oil in the long run, but I do not know of any prolonged tests showing that this is the case. I take it that the member refers to the gudgeon bearings of a trunk piston engine, and not to the top ends of a crosshead type of Diesel or semi-Diesel plant.

A MEMBER: In a great many cases, with the smaller type of semi-Diesel engine, you have frequently to place them in charge of a man who is more or less semi-qualified. He has not got a full qualification for the work, but we have to trust him. In this way a great many of the troubles which have occurred, particularly with the semi-Diesel engine, have not been due to the engine itself. I was interested to note the remarks in regard to generous lubrication where the author refers to the engine actually running away on the lubricating oil taken up from the crank chamber; I knew of a case of this kind, and I mentioned it to one or two people who might have been expected to have had experience of it, and they laughed at the idea. But it happened, as a matter of fact, on a trial of a submarine engine on the test bed in the works where it was being built. The engine had to be stopped for some purpose, but the thing would not stop with the fuel shut off. It was running entirely on lubricating oil. In that particular case, when the operating costs came to be worked out, the lubricating cost was heavier than the fuel cost, taking into consideration the quantity of each and the relative cost per gallon. On the subject of cylinder lubrication of these semi-Diesel engines, and of Diesel engines generally, the author refers to the method of supplying oil to the cylinder liners. It seems to me that the usual method of supplying oil to a chamber or passage round the cylinder and taking it away at more than one point round the circumference is not a good method of securing effective lubrication of the cylinder. The pistons, of course, are a good fit to the cylinder, but almost in every case it will be found that the piston lies more toward one side of the cylinder than the other, particularly where there is no crosshead, that is, where the gudgeon pin is in the piston. The oil supply escapes at the freest point and that is the point that requires least lubrication. The point where the piston is rubbing close to the cylinder wall is where it requires oil most. I would prefer to put a separate feed separately controlled to each of the points round the cylinder and reduce one and increase the other as required. I think that would be a much more satisfactory way of lubricating a cylinder. When the oil is supplied to the cylinder by the general system, oil is in the chamber or passage round the cylinder all the time. I think a much better plan would be to inject the oil at a certain time. I would put it in when the piston is at the bottom of its stroke, and the oil would then be carried up the cylinder walls by the piston on its return stroke. This is done by a good many makers now, and in every case it has resulted in much better working, and in a marked reduction of the quantity of oil used. The author has referred to the use of graphite in conjunction with lubricating oil. Personally I do not like the idea of adding graphite to the oil. I would prefer to keep to a clean clear oil without any material in it which could possibly help to increase the cylinder deposits, which are sufficiently troublesome as they are.

He says, in regard to graphite: "As regards petrol and paraffin machinery, its use has been attended with excellent results, as it is unaffected by heat, and not only lubricates the piston rings, but assists towards filling-up minute blow-holes, scores, etc." That statement contains in itself the condemnation of graphite as a cylinder lubricant, as if it will fill up blowholes, etc., it will also fill-up the spaces behind the piston rings. I should prefer to use a lubricant that would not tend to increase the deposits. Separation of the graphite from the oil will always tend to occur even when the special type of graphite, known as "deflocculated" is used. This is a manufactured substance, which will remain better in suspension in oil than natural graphite, but there is no means of keeping natural graphite in suspension in oil, and as soon as any water or moisture gets into the mixture the graphite separates at once.

A VISITOR: As a chemist, I regret Mr. Warne's decision from a practical standpoint to leave out considerations of quality, viscosity, etc. I should imagine that quality and viscosity are of paramount importance whatever engines the engineer may design, or whatever system of lubrication he may adopt. It seems to me that he is very largely dependent upon the quality of the oil. Either he must obtain the correct quality of oil to suit his design of engine or else design his engine to suit the oil. Since the engineer is primarily concerned with the design of the engine he prefers to select the oil which will give the best results subsequently. Therefore, from a practical consideration, perhaps a few more moments might have been spent on this particular phase of the question.

Reference has been made both by the author and several speakers to the continued running of an engine, after closing down, on lubricating oil. In my own experience I have found it quite an easy matter to keep an internal combustion engine running on lubricating oil after the fuel was shut off, but only with a certain type of lubricating oil, generally an oil more or less unsuitable, although a moderately good oil may possess the same propensity if it has suffered decomposition in an enclosed engine. Once again quality arrests our attention. At high working temperatures and pressures oil is liable to crack and give rise to substances which are sometimes highly explosive. This point was referred to rather fully in a paper which I have read before the Diesel Engine Users' Association in 1917.

I am a little lost to know whether or not the author recommends forced feed lubrication. Perhaps he will be good enough to express his views on this point a little more fully. Nor do I quite understand whether he suggests that oil grooves should be entirely eliminated. Surely the use of oil grooves involves the principle of oil-bath lubrication, efficiency of which is well-known and well-established.

Then as to working temperature: that is undoubtedly far more important than is suggested in the paper. One has to realise that the viscosity of any oil diminishes with rise of temperature, consequently if the working temperature be left uncared for it might steadily rise and the oil decrease in viscosity until the critical point is reached when either the piston seizes or the bearings go. The rise in temperature after a certain point is reached becomes very rapid, possibly too rapid to be dealt with before trouble is encountered. If wide variations of working temperature are permitted it is essential that a special oil to meet the conditions shall be used, otherwise unnecessary fluid friction is introduced which should be eliminated. How far the experiments carried out at the National Physical Laboratory on the lubrication of worm gears assist in solving the problems of Diesel engine lubrication remain to be illustrated. The sudden drop of efficiency, quoted in the National Physical Laboratory's reports, occurs at a temperature which is coincident with that at which a sudden drop in viscosity occurs. This may be a coincidence, or it may be a clue to the explanation; I cannot tell without a closer analysis of the reports, and perhaps more experimental work. Why castor oil might be expected to give best results is not quite as apparent to me as it is to the author. I have had a considerable amount of experience of castor oil, both alone and in conjunction with mineral oil, and the more research which I do the more complex does the subject appear.

The use of graphite has its limitations, although it has some advantages in special cases. In a general way I do not advocate its use, but there are cases where graphite has proved its value. Its particular advantage appears to be at the moment of starting, so that directly the starting has been effected, its sphere of usefulness diminishes. With the particular subject we have before us I do not think that anybody would seriously advocate the use of graphite.

Reference is made to the use of low viscosity oil for new engines. As the term "low viscosity" is a comparative expression rather than an absolute, a little qualification is necessary. The modest reference to the selection of oil for the air-compressors might well have been amplified by reference to the explosions which have occurred with the loss of life through the use of unsuitable lubricants.

I presume that the viscometer of simple design is the cup and ball variety, if it is I can say that my experience with it is not altogether a happy one; in fact I am unable to obtain even moderately consistent results, perhaps my instrument was not in perfect order.

Then, as to the use of waste for wiping out crank chambers. Periodically there is an epidemic of complaints of the oil containing fibrous matter from engineers, but on investigation the microscope soon reveals the exact nature of the fibre, and it is usually easily traceable to the wipers which have been used.

The subject of filtering the used oil has been raised, and I cannot do better than refer you to a general discussion on the recovery of lubricating oil, about two months ago, by the Diesel Engine Users' Association, in which I was invited to take part.

A Member: Viscosity to my view is a most important thing in regard to the oil for the engine, and a test for viscosity some engineers seem to fight shy of, but by practice it becomes quite a simple matter. I have used Redwood's viscometer many times to check the viscosity of an oil, and from my own experience I should have approved of the author of the paper enlarging upon the use of the instrument and advocating all engineers to become skilled in its use for checking the viscosity of the oil supplied. The subject can be enlarged upon so much that much time might be occupied in discussing it. The viscosity of an oil varies according to the temperature, and this is not always recognised; I would commend the study of it to all, also the testing of the oil. Referring to the stoppage of the engine mentioned, the oil we were using was very good oil, specially suited for the job. We used to draw the oil off. There was a steam coil to heat it, and we filtered it, yet the engine stopped. We overhauled it at Liverpool, and after the examination all we could conclude was that something must have got off the bearing and into a vital spot to stop the running. One thing I noticed, when using this particular oil was that the far away bearing got the least oil. For proper lubrication you must have all bearings getting their proper amount of oil, and there must be some means of seeing that the oil goes where it ought to go. The oil must go through its proper channel, and each bearing must have its proper quantity and this is sometimes a difficult thing to bring about.

A **Visitor**: The last speaker has suggested that all engineers should have the viscometer and flash-point apparatus. It is a comparatively easy test to those who are regularly engaged on the work, but even qualified chemists send out varying results from time to time with the ordinary viscometer. When you have these viscosities it is very handy.

The **Chairman**: That, I think, was what the speaker was aiming at—that there is room for more scientific investigation by marine engineers than at present. We could get a lot of information if the necessary instruments were supplied.

SESSION 1920-21.

President: LORD WEIR OF EASTWOOD.

Notes on the Management of Marine Diesel Engine Installations.

BY HOMER McCRIRICK (Member).

Read Tuesday, December 21, 1921.

THE Diesel Engine which has now established its position as a reliable and very efficient power unit suitable for ship propulsion does not seem to be getting the publicity from a practical point of view among marine engineers that it should.

The construction of large vessels which are to be propelled with Diesel Engines is increasing at a steady rate. Up till a few years ago developments with this type of engine were confined to land practice, and so the marine engineer is placed at rather a disadvantage as he had very little opportunity of gaining experience in this line during his marine training. At present most of the engineers for the new motor vessels will be men whose past experience has all been with steam plant, they will be seeking out what practical information can be got as to how the Diesel engine behaves, what troubles they are likely to encounter on long voyages, and the best means of remedying same before a serious breakdown occurs.

As the wading through books and technical journals is rather a heavy task when one has not too much time at his disposal, and as very little has been published regarding the practical side of Diesel work, a few notes from a marine engineer's point of view may be the means of opening up other sources of practical information. Many valuable papers have come before the Institute on the internal combustion engine, and much serviceable data can be got from a close study of same, but the present call seems to be from the more practical side. These notes which have been collected during the past few years both ashore and afloat are penned with the hope that they will initiate a discussion among the members and be the means of bringing the practical side of the heavy marine internal combustion engine to the fore among marine engineers in general.

The most successful and reliable type of Diesel engines at present afloat are working on the four-stroke cycle. For long continuous running when the power units per cylinder are over 100 b.h.p., the use of blast air for injecting the fuel into the cylinder is well worth the extra cost of power and extra space required for fitting compressors for this purpose. The little extra labour necessary for the upkeep of these compressors repays itself in the much longer time exhaust valves, fuel valves and pistons can remain in operation without being dismantled for overhauling. This saving of labour can be put down to a more complete burning of the fuel taking place in the cylinder, also I think it can be said that the extra number of heat units obtained from the fuel for useful work in the engine when blast air is used will compensate for the power required to drive the air compressor.

In two-stroke engines which exhaust through ports that are uncovered before the bottom dead centre, trouble seems to have been experienced with the water jacket and liner joint near these ports, also the cracking of the liners caused by the unequal expansion and rather awkward construction in way of the ports. Against this we have no exhaust valve troubles, but an exhaust valve is much easier renewed than the water jacket and liner joint. Pistons in the two-stroke engines seem to have given more trouble than the pistons in four-stroke engines owing to overheating and cracking, as they have twice the amount of heat to dispose of in a given time. Piston trouble seems to be now receiving marked attention from the metallurgical side, and this, combined with the various reliable piston cooling systems now in operation, should help matters greatly, also with careful attention by the ship's engineers to the fuel pump adjustments, cylinder lubrication and the piston cooling water temperatures, the Diesel piston should not require any more overhauling than the high pressure steam engine piston.

When a piston becomes badly cracked the cylinder gases will pass through to the water side and displace the cooling water from the piston, this can be detected by only hot gases passing from the piston cooling return pipes, and it is advisable to shut the fuel and blast air off from that cylinder and have the piston removed at the first opportunity.

Cylinder Heads or Covers. — In the four-stroke engines, cylinder heads or covers are rather intricate castings owing to the number of valves which have to be accommodated in them, namely, fuel, air starting, exhaust and inlet. Many points have to be considered in the design of the cover, its large depth to allow of sufficient area for the air inlet and exhaust gases passing to and from the cylinder. The circulating water must have a clear passage over as much bottom area as possible. Provision has to be made for a passage through the water space to carry the starting air to the starting valve which will be at a pressure of 20 to 25 atmospheres. The air passage which has to carry this high pressure will most likely be eliminated in the future, and the air starting valve designed to take air from a pipe coupled up to it on the outside of the cover.

The under-side of the cover when the engine is working will have to withstand a pressure of 40 atmospheres when the average temperature during the power stroke is about 2,200° Far., so any bad distribution of metal is liable to cause distortion, and cracks will probably result. Care should be taken that no large quantity of sand or mud is allowed to find a way into the covers and deposit on the bottom. Hand doors are provided for the examination of the inner surfaces, and should be opened occasionally, and any deposit removed. Deposits are also likely to be found at the bottom of the liner water jackets, this is not so serious provided it does not choke up the water passage. In small engines of less than 100 b.h.p. per cylinder it will be found advisable after three or four years operation to have the cylinder liners drawn, in most cases the small ports round the top of the water jacket will have become partly choked and will therefore be liable to cause over-heating to the covers through insufficient circulation.

Many cracked cylinder covers are at present doing duty on engines running with full load; if much manœuvring has to be done these covers become a source of danger, and whenever a stop is called for any length of time it is always advisable to shut down the circulating water pressure from the engine and so minimise the risk of water finding its way into the cylinder in case the covers have time to cool down and perhaps the cracks open. These cracks usually extend from the fuel valve across to the inlet and exhaust valves, round this area is where the water passages are greatly contracted and a most likely place for mud to deposit.

The joint between the cover and liner is usually a ground face to face joint with a little oil and graphite painted on the surface. When these joints show signs of leakage it is advisable to have them seen to at once, as the passage of hot gases between the two surfaces will soon burn the spigot. Grinding rings are sometimes supplied for facing up these joints or else copper joint rings can be inserted between the faces. Cases have been known where the burning has become so deep that a piece of copper had to be dovetailed and pinned into the spigot before a tight joint could be obtained.

Fuel Injection Valves.—One of the characteristic parts of a Diesel engine is the fuel valve, the whole success of burning the fuel properly lies in the design and adjustment of this valve. With regular overhauls and the use of a good packing material, very little trouble will be experienced, the gland packing box which has usually a depth of four spindle diameters seems to answer its purpose well when packed with good lamp cotton plaited to size and well soaked in oil and graphite. There are one or two kinds of semi-metallic packing on the market, which also answer their purpose well, but what must be guarded against if possible, is wear taking place on the spindle where it passes through the packing, this wear, which is often found to start in the form of pitting, may be caused by some chemical in the oil. After it has been going on for some time slight scores or grooves form along the rod in way of the packing, this I think can be put down to a fine grit finding its way into the packing from the oil, and though the spindles are case hardened, it seems to have effect on them. It is always advisable when overhauling fuel valves to try and smooth over any roughness on the spindles with a very fine emery stone.

The most destructive trouble with the fuel valve is a small leak starting on the valve face, this may be caused by a particle of hard grit adhering to the face and allowing burning to take place thereon. A fuel valve with a slight leak caused by the valve face being pitted or burned, or the spindle partially seizing in the gland, can very often be detected by a slight blow back of air through the inlet valve muffler during the time the inlet valve is open.

If a leak attains any size, dangerously high pressures will be formed in the cylinder causing the escape valves to lift. Pre-ignitions are liable to occur with a leaky valve and they can be detected by a dull heavy knock. A fuel valve leakage caused either by a burned valve face or a seizure of the valve spindle, is quite capable of bringing the engine to a stop through loss of blast air.

Seizure of valve spindles mostly occurs when starting up a cold engine or when manœuvring after a long run. Sometimes this sticking of a valve spindle can be overcome if sufficient blast air can be obtained for the short time of manœuvring the engines, but it is not recommended that this be done unless in cases of urgent necessity, as there is always the risk of dangerous explosions occurring in a cylinder with a leaky valve.

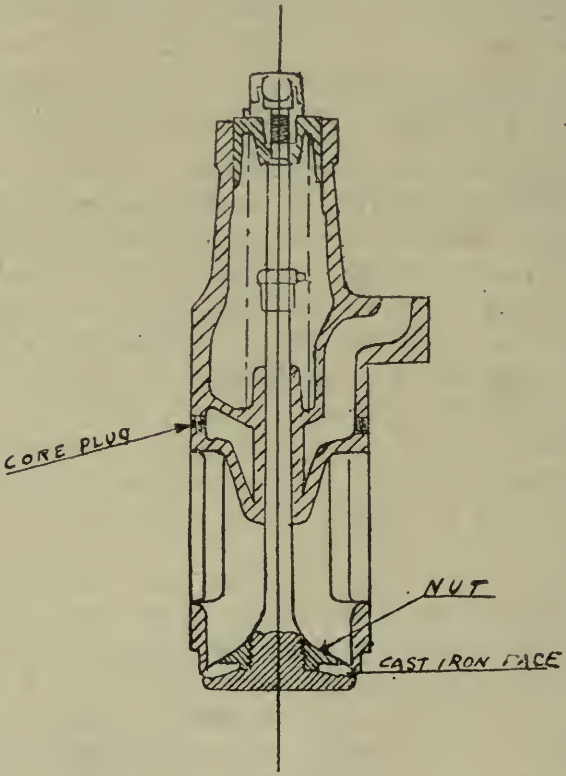
The pulverising and spraying of the fuel is effected by means of a number of drilled or slotted washers and a special grooved cone piece. Generally speaking, these do not receive very much attention apart from cleaning when the valve is opened up for overhauling. At the present time most motor ships are on regular runs, and so able to obtain the most suitable fuel oil, these conditions are not going to continue for ever with the increasing number of motor ships. More practical information is therefore wanted for the adjusting of pulverisers which will enable the engineers to judge different grades of fuel oil and so meet any valve trouble before it becomes serious.

Another rather important item in the fuel valve is the small non-return valve fitted in the valve casting to prevent blast air finding its way down to the fuel pump. These valves do not require very much attention apart from an occasional grind on to their seats with very fine powder, care must be taken when the small valve cage is replaced in the fuel valve casting that the copper ring joint is well bedded all round. As a further precaution against blast air passing the non-return valve and finding its way to the fuel pump another valve of the same type is inserted in the fuel pipe line. When only one non-return valve was fitted to each fuel valve, a leakage of air past same, caused by either the valve sticking open or a piece of grit adhering to its face, would be liable to cause a stoppage of the engine if the air passes in sufficient quantity to displace the oil from the pump.

Exhaust Valves.—Valve faces in exhaust valves are manufactured from a special mixture of cast iron suitable to withstand gases at a temperature of 700° Far. passing over them. Valve spindles are made of steel and means are provided in large valves to renew the cast iron faces on them by unscrewing a nut and bedding on a new C.I. face.

In exhaust valve cages the spindle guide is usually water-cooled as shown in Fig. 1. After these valve cages have been in use for some time corrosion has started at the core plugs and allowed water to find its way into the exhaust piping. If the engine has been standing for any length of time with this leakage going on, a considerable amount of water may find its way into a cylinder through an open exhaust valve. The trouble is then to find where the water has come from, as in large

FIG. No. 1



engines there is always the possibility of a leakage starting in the water-cooled exhaust box, or cylinder covers may be cracked and leaking.

Broken levers have been caused by springs being fitted into valves and not having sufficient compression clearance; whenever a new spring is fitted or an old one packed up with washers always make sure that the valve has the required travel.

Inlet Valves.—These valves are practically of the same construction as exhaust valves, except for the omission of the removable C.I. valve face and the water-cooled spindle guide. Air strainers or silencers are fitted on the inlet induction pipes, these must be frequently cleaned to allow of a sufficient supply of air reaching the cylinder during the time the inlet valve is open and so permit the complete combustion of the fuel. If this condition is not fulfilled a vacuum is likely to be created in the cylinder and the slight back pressure of the exhaust gases will tend to jump the exhaust valves off their seats and allow burnt gases to pass back into the cylinder, this will result in bad combustion, loss of power and a smoky exhaust.

Air Starting Valves.—These valves are only in operation when starting up the engine either ahead or astern; immediately the engine is put on fuel the starting valves are automatically cut out of action. When engines are slow in picking up on fuel some engineers are inclined to interfere with this automatic arrangement and allow fuel to be injected into the cylinder when the starting valves are also in operation. This practice is not to be recommended as when the burning of a fuel charge takes place the pressure will be above the starting air pressure, and if any oil should have found its way into the starting air system there might be the danger of an explosion in same. There is also the danger of burnt gases carbonising the starting valve spindles.

The starting air enters the valve by a passage through the cover, the pressure tending to open the valve is balanced by the large diameter of the spindle above the opening; in large engines this part of the spindle is fitted with small Ramsbottom rings to ensure tightness. Starting valves should receive the same careful attention as fuel valves, as in all likelihood a starting valve sticking open a little will prevent a start of the engine until it is freed. A slight leakage past the face of a starting valve while under way can sometimes be overcome by a restart of the engine with air, or by turning the valve round to a different position on its seat. If this will not stop the leakage it is advisable to change the valve at once as the small hole or groove will only enlarge with the passing of the burning gases. As the valves are common to the one air pipe, these gases, if allowed to pass, will only tend to carbonise up the starting valve spindles. The cage for holding the starting valve in the cover has usually a metal to metal face joint at the bottom, and provision is made for a thin soft joint at the top. The space between these two points is where the air is led to the valve; care should be taken that both joints are properly bedded, as a leakage at the bottom would admit air to the cylinder when not required, or the cylinder gases will pass back to the air starting system when the engine is under way. A badly made joint at the top would cause a loss of starting air, or with the engine working on load, any fuel oil leaking from the fuel valve spindle gland may find its way into the starting air system and might be the cause of an explosion the next time the starting valves came into operation.

Valve Timing.—In engines working on the four-stroke cycle the following table gives approximately the timing of valves in cylinder with the I.H.P. in each ranging between 100 and 400 and with a piston speed of 700 to 800 feet per minute:—

Fuel Valve opens 6° before T.D.C. and closes 34° after T.D.C.

Starting Valve opens 10° after T.D.C. and closes 140° after Top Dead Centre.

Exhaust Valve opens 40° before B.D.C. and closes 10° after T.D.C.

Inlet Valve opens 20° before T.D.C. and closes 15° after B.D.C.
The same settings are arranged for ahead and astern.

Great care should be taken when adjusting valve lever or push rod clearance on cams. After any large overhaul, or once a year, the fuel valve settings should be tried for correctness. In small engines where the cam shaft is driven by bevel or skew gearing from the crank shaft, and every part of the vertical shaft, coupling, and gearing is enclosed, any slackness of keys is apt to remain unnoticed. If a key has been an easy fit sideways when the engine left the works, it is bound to work itself further slack in a year or so, with the result that the cam shaft setting will be retarded. A key slackness of a little over 1/16th inch that came before the writer's notice, was found to alter the valve timing by 6°, when this engine first showed signs of sluggish starting it was put down to change of temperature conditions in the engine-room. After this passed over and there was no change in the starting up of the engine the trouble was looked for in the valves, fuel pump and cylinder compression clearances; as these were all found in order, a further search had to be made which resulted in a slack key being found in the muff coupling of the vertical shaft, which drove the cam shaft.

Indicator diagrams play a most important part in tuning up the Diesel engine, they provide the quickest means of ascertaining what adjustment may be required to obtain the best combustion of the fuel and also seeing that the load is distributed equally in each cylinder.

Fig. 2 is the approximate form of a theoretical diagram. Compression starts from A when the air will be at a temperature of about 120° Far., it then continues until point B when the temperature will reach 970° Far. and the pressure will be about 500 lbs. per square inch. The injection of fuel oil now starts and it is injected between B and C; combustion should proceed at a rate which will permit the increase of volume, due to the motion of the piston, to be so balanced by the increase of temperature, due to the combustion of the fuel, that the pressure will remain constant until the whole of the fuel has been injected when the temperature at C will be about 2200° Far., burning of the fuel still continues, it is stated, until the piston has travelled about one-fifth of the stroke, when expansion will

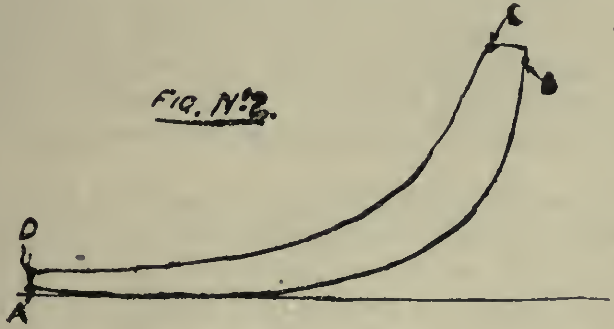


FIG. No. 2.

then continue until the exhaust valves open at D and the gases will be released at a temperature of about 750° Far. To come as near this ideal condition as possible in practice, all valves in the cover must be kept in good condition, rings must be free in piston, and the compression clearance should be adjusted regularly, as the wear on top and bottom ends is taken up.

When the engine has been running for a little and is properly warmed up, compression cards should be taken from each cylinder with the fuel and blast air shut off, everything being in order they will indicate a pressure of 500 lbs. per square inch. We now come to the combustion of the fuel, which should be sprayed into the cylinder in particles of such a size as will expose sufficient surface to allow of combustion taking place within the proper time, and also not of a very violent nature. If combustion takes place too quickly, the blast pressure should be reduced, as the greater the excess of blast pressure above that of compression, the more quickly will the quantity of oil measured out per impulse of the fuel pump into the valve casing be blown into the cylinder. If the fuel oil is of such a highly inflammable nature that a reduction of the blast pressure will not regulate the period of combustion sufficiently, then it may be advisable to alter the sprayer plates in the fuel valve and so prevent the fuel oil from being broken up too finely. On the other hand, care must be exercised when any adjustment is made of the sprayer plates, that the combustion of the fuel will not be delayed to such an extent that burning will take place during the later part of the expansion stroke.

The effects of after-burning have the same effect as incomplete combustion and cause excessive cylinder temperatures, which tend to burn the film of lubricating oil from the cylinder walls. This shortage of lubrication will cause increased wear on the cylinder walls, danger of piston seizure and piston rings sticking. In large engines with open cylinder ends, shortage of cylinder lubrication can usually be detected by a slight grunting noise from the affected piston. If caught in time, the supply of oil from the lubricator can be increased for half-an-hour or so, but if the piston has become much overheated and sparks are passing down from same it is advisable to cut off the fuel oil from that cylinder, slow down the engine, and attempt to spray cylinder oil up inside the liner with a syringe.

The combustion part of the cycle takes place when the piston is moving slowest, with the usual type of indicator gear supplied, the record of pressure change in the cylinder at this time is put into very small space. Should further investigation be required as to what is taking place during combustion, it can be obtained with a draw card, this can be taken by disconnecting the indicator cord and at the instant that fuel starts to be injected into the cylinder, the indicator pencil should be held to the card and the drum quickly moved round by hand.

In figure No. 3 a full power diagram is shown, the line above the top of the diagram is the usual type of draw card obtained when combustion is proceeding in the required way. At point A the injection of fuel is starting, a slight rise of pressure is indicated here and continues for a time until the fuel valve closes and the burning of fuel ceases and expansion starts.

FIG. Nº 3

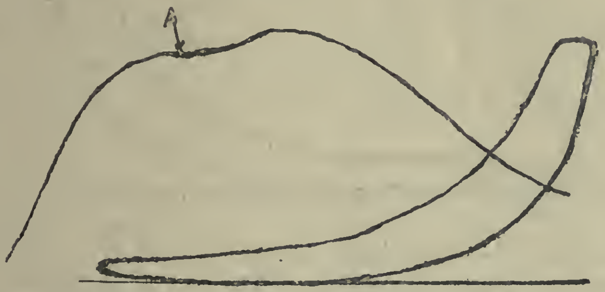
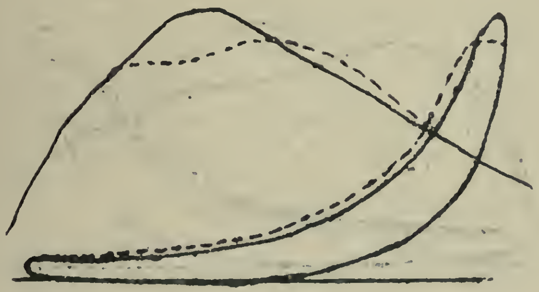


Figure No. 4 is a card from a cylinder having rather early ignition of fuel, the height of the card has increased and the

FIG. Nº 4.



burning of the fuel appears to take place quicker. This can usually be overcome by increasing the clearance between the push rod and the fuel cam a little, should a slight adjustment of the clearance not be sufficient then the position of the fuel cam piece will have to be put back a few degrees.

Figure No. 5 is a card from a cylinder having very early ignition, if the valve setting is correct this may be caused by a leaky valve or else the spindle is seizing to such an extent that the force of the spring cannot close the valve. A valve in this state will not continue to operate long, if it is not convenient to stop the engine and change the valve, it is advisable to shut the fuel and blast air off, as dangerous compression pressures may be generated in the cylinder.

FIG Nº 5.

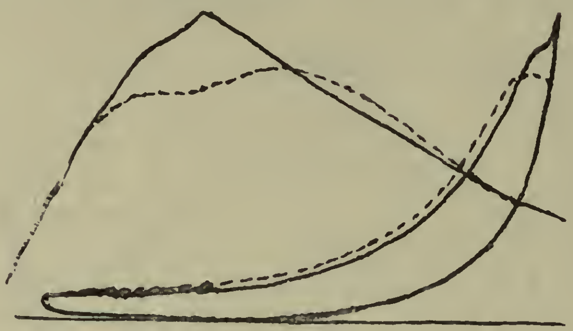
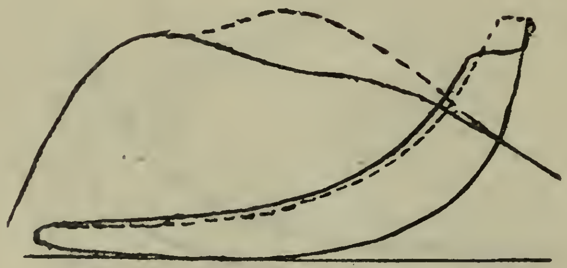


Figure No. 6 is a card from a cylinder having very late ignition. In cold weather, cylinders with the ignition taking place

FIG. Nº 6.



so late will be difficult to start firing when getting under way or manœuvring, and the combustion of the fuel will not be perfect, resulting in a smoky exhaust. A card of this type may also be obtained from a cylinder using heavy oil which has not been pulverised sufficiently, and also through using too low a blast pressure if the engine is on load.

There is also the possibility that sufficient air is not finding its way into the cylinder to allow of proper combustion. If there is any doubt about this, a card taken with a weak spring in the indicator will indicate by the distance of the exhaust or inlet above or below the atmospheric line whether there is too great a resistance for the gases to leave the cylinder freely, or the air to enter freely.

At light loads there will be more than the required quantity of air in the cylinder, and with the extra air coming in as blast along with such a small quantity of oil, the ignition is bound to be retarded owing to the cooling effect of the large proportion of blast air over the fuel.

Particular attention should be given to the fuel pump adjustments, once these are set and a card taken from each cylinder showing the required mean pressure, which is generally about 6·5 Kg./C.M2 or 95 lbs./sq. inch, the fuel pumps should not be individually interfered with to increase the quantity of oil supplied to any one cylinder which may show a falling off in power during continuous running. The trouble will in all probability be other than the fuel pump adjustments, and may be such as inefficient supply of air to the cylinder caused by either a partially choked inlet or exhaust, or a little grit interfering with the operation of fuel pump valves. The first cause may be removed when overhauling at a port of call, the second may remove itself any time during the ordinary running of the engine. If the amount of fuel delivered per stroke of a fuel pump has been increased over the ordinary settings of the other fuel pumps, and the loss of power has been removed from the cylinder by causes as stated above, this cylinder will now start to develop much more than the required power and a cracked piston may result before the increase of power is noticed.

Air Compressors.—Air compressors are of the two stage trunk piston type in small engines and the three stage type in large engines. In the past, most of the trouble with compressors was due to valves breaking; a large amount of attention has been given lately to the design of these valves and also the quality of material used in their construction. Valve trouble has now been reduced to a minimum, and compressors can run for months without any overhaul, provided the lubrication of them is carefully watched by the engineers in charge. Excessive lubrication seems to be responsible for most compressor worries at the present time, the surplus oil gets carbonised and adheres to the valves, with the result that through time they become choked up.

The air strainers or silencers on the L.P. stage must be kept free from dirt and oil, so as to prevent the slits in them becoming closed up and not allowing sufficient air to pass into the compressor. If the passage of air is any way restricted, a vacuum will be formed during the suction stroke and in a two-stage compressor of the trunk piston type there will be a tendency for vapour and oil to pass from the crankcase to the top of the compressor piston. Scraper rings are fitted in the lower end of the piston to prevent this, but unless these rings are kept a good fit the lubricating oil seems to find its way past them, and will eventually deposit at the valves, or in the air passages, and become a source of danger.

The safety valves on the water side of the intercoolers should receive periodical attention to prevent their sticking fast, in case a leaky tube or coil tends to raise the pressure inside the water casing to a dangerous height.

The air passages which are part of the compressor casing containing the suction and discharge valves at each stage, are also fitted with safety valves, which should receive occasional attention, as there is a danger of explosions taking place in these passages, through excessive lubrication and the heat of compression in the air.

A small air leakage from a compressor coil or intercooler may be detected when the engine is running, by the pumping action of the circulating water when it leaves the ship's side. Or by leaving the cooling water circulating round the compressor or intercoolers, when the engine is stopped any leakage will show up by a continual drip of water from the drain on the air side of the coolers.

A quantity of lubricating oil seems to find its way into the L.P. and I.P. coolers with the compressed air, and adheres to the tubes, reducing their cooling effect, these tubes are so arranged that they can be bodily withdrawn from the cooler casing for cleaning purposes, and under ordinary running conditions it will be advisable to clean the tubes about every two years.

The greatest wear in compressor cylinders takes place at the top end of the H.P. stage, and after three or four years running it will be found advisable to have them bored out parallel and a new top fitted to the piston.

Lubrication.—The general practice now with large engines is to supply oil under pressure to all the main bearing surfaces, small bearings, such as cam shaft and levers, are oiled by hand.

The crank cases are entirely closed except for small breathing holes situated near the top. In the large 3,000 I.H.P. installations, about 20 tons of lubricating oil is carried in a small part of the cellular double bottom, this oil which is circulated round the engine bearings falls to the bottom of the crankcase and is then drained to the double bottom tank, where it has the benefit of the cooling surface of the ship's skin.

The lower ends of the cylinders in large engines are usually separated from the crankcase by a distance piece, fitted with a tray through which the piston rod passes. This space between the lower end of the cylinder and the top of crankcase is open to the engine-room, and means are provided to drain away all the surplus oil which gathers here, this waste oil should on no account be collected and used for lubricating purposes, being mainly composed of oil that has had its substance burned away, and containing particles of carbon and ash that have been carried down from the piston top, it will thus act as a fine grinding paste on any bearing surfaces that it comes in contact with.

Careful attention should be given to all connections and joints in the water-cooled guides and piston cooling gear, so as to avoid any large mixing of water with the lubricating oil. After any large overhaul of moving parts, any waste, wood, or other refuse should be carefully removed from every corner of the crankcase, or they will only become a source of trouble when the engine gets under way. In cold weather, lubricating oil that has had a large percentage of salt water mixed with it, will tend to thicken up, and trouble will most likely be experienced with the pumps and also in getting the oil to flow freely back to the D.B. tanks. This may cause serious delays and a lot of unnecessary work and worry on the part of the engineers, so the importance of keeping the lubricating oil free from all foreign matter cannot be too strongly impressed on all the engine-room staff.

The consumption of lubricating oil with 3,000 I.H.P. installations under full running conditions, will average about 1/14th of a ton per day of 24 hours, any loss can be put down to evaporation and small leakages. To keep the oil in as good condition as possible, I think it is advisable once a year to allow it to settle for four or five days and permit any foreign substance or water to fall to the bottom of the tank. The suction pipe should then be raised about eight inches above the bottom of the tank, and the oil pumped back into the crankcase, the tank can then be thoroughly cleared out and fresh oil added to make up the deficit.

Engine and compressor pistons are lubricated by small force feed oilers, which are filled up every watch by the attendant; in view of the importance of compressors being supplied with the correct oil, it will be found expedient in large motor ships to paint compressor oil tanks, oil can and lubricants, a distinct colour, and so avoid any errors in oiling when perhaps the vessel has started on a voyage, with a new engine-room crew.

Lubrication troubles have been caused in auxiliary engines by the oil adhering to the walls of the oil passages in the crankshaft and eventually closing them up. When any serious oil thickening has been encountered, a little time spent blowing steam or compressed air through the crank shaft oil passages, may save many hours work at some future time, and also the cost of replacing overheated bearings.

With care and good judgment on the part of the ship's engineers in the adjustments of the motors, a well kept storeroom and properly indented spare part list, there is no reason why the Diesel engine cannot be made as reliable as steam plant. Repair bills on the old ships seem to be on the high side at present, but with the increasing improvements in design and choosing of suitable material for the various parts which show deterioration, the bills will be greatly reduced in the future.

NOTES ON THE MANAGEMENT OF MARINE DIESEL ENGINE INSTALLATIONS.

DISCUSSION.

The Chairman: It is only my part to allude to one or two salient points. I want to know, first of all, the differences in the expressions of opinion by the writer and the author of the paper published in our last edition of the journal. The last edition contained a paper on solid injection. The author of this paper deals more with the air blast system. There is no mention made in this paper of the opposed piston type, now receiving a good deal of consideration. Some information in regard to that kind of engine would be of very great use; also on the relative values of the four-cycle and two-cycle types. The author seems to think the four-cycle is the more reliable. There are some authorities of considerable standing who think differently. They think, at least, that whatever the defects of the two-stroke type may be at the moment, the two-stroke will ultimately become the predominant type at sea.

A Member: In the case of the plain Diesel engine I think a great deal can be done to assist the main running by due regard to the oil filters. In a Diesel engine of 300 or 400 h.p., carbonised oil may be dropping into the crankcase, and often the filter is placed in the bottom, not readily come-at-able and it is not convenient always to stop. To quote one instance, we found it much more convenient to put a by-pass into the suction line and have a filter similar to a bilge mud-box, which was of great assistance to us in cleaning out this carbonised oil. Another point in regard to the large type of Diesel engine was that we always made it a practice when over-hauling in port to clean out the crankcase thoroughly, put on the oil pressure from the main and examine every bearing to see the oil was going to each. As a good deal of pressure is often lost by a badly fitting liner these ought to be watched. We checked the oil system before starting up, including the oil boxes, both for the lubricating and pressure oils. Each trip we had probably different men; this caused trouble in training the attendant to watch and note where the different oils should go. In order to lessen the risk of mistakes in the oil supply system, the various oil boxes should be plainly marked. We found great help from having a guard fixed to the doors along the cross crankcase, as it prevented a lot of the carbonised matter dropping down, and after the engine had been running for some weeks we found the guard covered with carbonised oil. The course of the oil can be checked at the sight feed, but often you are not sure that it is getting ample.

The CHAIRMAN: This type of engine will clearly be developed by men with years of sea experience behind them. There is no doubt that the Diesel engine has a great deal to be improved before it is thoroughly established, but it has proved so efficient that now its position is unassailable. We know, as practical engineers, that the mechanical difficulties and the details of design have yet to receive much attention before they approach the degree of perfection arrived at by the steam engine. The reason is that the work hitherto has been developed on divergent lines.

We have a small number of large firms who have developed this engine on lines of their own, so much so that amongst some of the first class people we find very strong divergencies of construction. The mechanical perfection which we have evolved in the steam engine took a long time to achieve. It is practically perfect to-day. Whoever builds a quadruple expansion reciprocating engine to-day you will find it is pretty well the same from whichever works it comes. The same process will have to be carried through with the Diesel engine. It has to be standardised. Ultimately what is best in each firm's design will be embodied in a design which will eliminate the more difficult and intricate features, and at length we shall have a type of engine which will be as simple as it can possibly be made, and which, at the same time, will preserve all the advantages which have hitherto been evolved by the best brains given to the subject. I think that Diesel engine designers and those in charge of Diesel engines will learn a great deal from those who have had a long experience of the steam engine. The steam engine which appears so familiar and so obvious to us, took a long, long time to perfect. I notice that a writer, very recently, has stated that the differences in mechanical design which exist between the Diesel type and the reciprocating steam engine will in process of time disappear. In other words, the Diesel engine will tend to be more and more modelled on the lines of the steam reciprocating engine. Take the question as to whether the engine should be of the enclosed or open type. Can you suggest any reason why the makers of Diesel engines should continue to adopt the enclosed type of crankcase? Similarly, on the question as to whether the opposed piston type, on the two-stroke cycle or the four-stroke type, with fixed cylinder head, has most in its favour.

A MEMBER: The author states in one paragraph: "The safety valves on the water side of the intercooler should receive periodical attention to prevent them sticking fast, in case a leaky tube or coil tends to raise the pressure inside the water casing to a dangerous height."

I quite agree with him, but let us proceed further. The intercooler castings are tested to about 100 lbs. per sq. in. The coils inside contain air varying from 300 to 1,000 lbs. sq. in. Two accidents with intercooler coils came under my notice when at sea on a Diesel-engined vessel. The intercoolers were fitted with the usual safety valve. The valve was adjusted to 25 lbs. per sq. in., and it was about the same area as the circulating water discharge pipe. One intercooler contained a coil at 300 lbs. pressure, the other a coil at 900 lbs. pressure. In each instance when the coils burst the intercooler casting burst also, despite the safety valve. I have heard of similar accidents, and it appears to me that something more positive than a safety valve is required on the intercooler. I would suggest that a disc, similar to those found on the CO_2 machine, be fitted about 4 or 5 inches diameter, so that in the event of a coil bursting the disc would blow out, save the casting, and probably avoid fatalities.

With regard to starting valves, anyone interfering with the automatic arrangement and allowing fuel to enter the cylinders with the starting air is courting disaster, an explosion would probably occur and wreck the whole engine. A member mentioned an incident which did happen when fuel accidentally found its way into the starting air system.

With reference to the paragraph on cards: It is not always convenient to take compression cards, in the method mentioned, off the "mains" after starting up; further, there is not always time to spare for this purpose. My practice was to obtain a set of draw or hand cards at first opportunity, and from these could always assure myself whether the cylinders had proper compression or not, and then adjust accordingly.

The author states that the average temperature during the power stroke is 2,200°F. I should be glad if he would state at what mean effective pressure the engine referred to worked at, as it appears to me a very high average temperature.

The CHAIRMAN: This is shown as a card indicating a wrong condition in the engine. If everything is working all right you get your ordinary compression right from the ordinary card, and if you do not get your compression right, it is an indication that there is something wrong with the engine. The remarks on the subject of the safety valve are really weighty remarks; two instances have been quoted of accidents that might have been fatal, and in one case did result in injury. It is difficult to say how far the present practice in regard to evaporators applies. It is precisely that state of affairs that we are trying to guard against in the case of evaporators to-day. We have a diaphragm with an orifice and the area of the orifice is calculated to be well within the danger mark. The fitting of a disc would be a matter of some nicety, that is to say, a disc of sufficiently large an area so attached that there would be no fear of it blowing off at normal pressures, whilst yet it would be certain to blow off when called upon to handle a very much larger pressure. Perhaps the previous speaker would suggest the best way in which that disc could be attached. I think this is very important, and I think it will be one of the most useful contributions to the discussion. I will ask him to give his opinion as to how that disc can best be attached to be firm whilst only dealing with water pressures, but be sure to operate if called upon in the case of explosion or fracture.

A MEMBER: In reply to the Chairman the disc could be fitted in the form of a blank flange, with a compensating ring on its outer edge for stiffening to allow for jointing purposes, the diameter of the disc to be about 5 ins. I think that would be quite effective for the compressors on board any of our large motor ships of the present day. The disc could be made of some thin metal; copper would serve.

The circulating water pressure is usually about 20 lbs., seldom more. The disc could be designed and constructed to operate at 22 lbs. so that in event of the cooler casing attaining that pressure the disc would blow out.

When a coil bursts, the air impinges on a solid mass of water, something must give way, and it is usually the casting. That is why I recommend a disc which would be weaker than the casting, and could be easily replaced at small expense.

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A Member: In my experience we had 300 lbs. on the square inch and the liability of explosion. The relief valve could lift up at the arranged pressure. The disc put in here was of lead, 4 ins. diameter and 5/16 in. thick. There were four or five vessels connected with this SO_2 arrangement. This was the master valve in the whole lot to prevent explosion.

In the event of a defect or a big pressure, this disc would simply break and save the whole system all round so that no damage would be caused in that direction. The idea is that this disc rips open, and you can easily replace it. That is the style that the valve can be made so as to be quite safe. A member has referred to an explosion. I have not previously met anyone who has had an explosion experience, and it is a point I have gone into once or twice—the possibility of an explosion with the two and three-stage air compressors. Can you tell me: Have there been any gases formed on the walls of the cylinders of the air receivers? Is it from the lubricating oils, or is it a question of simply allowing the air to condense down and create this air liquid? There is drainage of course. When you close down you get a drainage. Does it carry any hydrocarbons and these turn into a gas to attack the steel of which the shells are made? We have a lot of pitting, and this wasting by pitting is a source of danger to the engineers in charge, whatever the plant may be, whether on board ship or ashore. The question is, "What is the cause of this deterioration going on?"

A Member: The air and carbonised oil, etc., carried with it, travels at high speed through the coil, and it has a scouring effect in the coil. On one occasion I opened out a burst coil for examination, and found that the inside was oval, the inner half retained its original thickness, whereas the outer half had worn quite thin.

A Member: With reference to the insurance of Diesel engines, it is usual to have the air compressor copper coils weighed when first put in, a record of this weight being kept. The coils are weighed again from time to time during overhauls. When the weight is reduced about ten per cent. the coil is rejected as being of no further use and a new coil is fitted. The acid in the oil deteriorates the coil, as well as the abrasive dust in the atmosphere. The oil used in the main cylinder is very often not good for compressors due to cracking and formation of acid at high temperatures.

With reference to preventing fracture of air-cooler water jackets, instead of the copper suggested, vulcanised rubber discs have been fitted and they have worked efficiently.

I do not know how it would serve at sea, but for land work it is all right. The author refers to cracked cylinders and cracked pistons and the shutting off of fuel and blast air, but you are very likely to get water into the compression space if you do this. The piston comes up to within about five-eighths inch of the head of the cover, so if you get water into the compression space you will get a damaged connecting rod and very likely wreck the whole engine. Some submarine engines were damaged in this way. Our practice was always to turn the engine to get rid of the water before starting up by means of small compression cocks or indicator cocks placed on a level with the top of piston when on top centre. Most station engineers now turn their engines before starting up to get rid of the water.

In regard to this question of the fuel supply through the Diesel valve, I do not think you pay sufficient attention to the fact that this valve is only open for about one-sixteenth of the stroke, and it is highly important that this period should be absolutely the same for each cylinder, and if the clearance between the Diesel valve and the cam varies the slightest amount, the cylinders will be unequally loaded. I refer to the case where you have got a distributor instead of a separate pump per cylinder, when each cylinder has a separate fuel pump great care has to be taken to insure that all pumps are delivering the same amount of oil as each other, in proportion to the load on the engine.

It is impossible to take compression diagrams with fuel and air supply on. I do not think you could possibly tell what compression you have unless you shut off the fuel and air, because you might have a false compression due to a leaky Diesel valve.

In regard to shutting off cylinders, is it advisable to shut off too many cylinders? You are liable to get a strain on the crankshaft due to too few cylinders doing the work. A lot of trouble in land stations is due to strained crankshafts owing to uneven turning moments and excessive pressures. Great care must be taken to keep the crankshaft dead true in its bearings, as constant flexure due to uneven support causes crystallisation and ends in a broken shaft.

The CHAIRMAN: The last speaker has given us some very useful information in regard to obtaining the compression temperature without cutting off the fuel. I take it the speaker will agree that that method would only be operative when the ignition was normal. In Fig. 5 in the paper we notice that there is an altogether artificial height on the card which is due to pre-ignition. I take it that in that case a card of that description would be no indication of the actual compression in the engine itself.

A MEMBER: In one part of his paper the author refers to the fact of many engines running with cracked cylinder covers, by which I take it that although known to be cracked they are not renewed. This appears to me very strange, as the author previously informs us that the pressure in the cylinder at times reaches 40 atmospheres.

The CHAIRMAN: You will understand this cylinder cover is hollow. There is a top and bottom, with a considerable depth between. If the under surface is cracked the flange and upper portion, which take the thrust, are intact. The action, referred to in the loss of weight in H.P. compressor coils, is really due to a peculiar type of erosion. It is not corrosion. It is erosion of these pipes due to the passage of hot air at a high pressure, and at a high velocity, and it is a fact that some very serious accidents have occurred in these intercoolers. These pipes, which convey the air, work in a chamber surrounded by water, and these pipes have exploded in a way that in the beginning was quite inexplicable. It was only the adoption of the ingenious idea of weighing these pipes that has enabled those in charge of these plants now to satisfy themselves as to the degree to which wastage has gone. That point I wanted to emphasise, in case it might not be apparent to those who heard of it for the first time precisely what was meant.

The scarcity of oil is a rather serious setback for this type of engine at the present time. That scarcity of oil will probably be overcome in one way or another. If we cannot find our oil, we will grow it in the fields. It is perfectly feasible and without a doubt we can expect ultimately that large tracts of the earth's surface, which now bear no crops could be adapted to the growth of root crops specially raised and developed for the purpose of bearing a large alcohol yield. That is one of the lines upon which internal combustion engine fuel will probably develop. We know that the oil problem at the moment is a difficulty; but it is not permanent.

A Member: The Diesel engine has come to stay, and the modern marine engineer will no doubt be brought in contact with it sooner or later, so without question, it is well worth considerable study. There have been difficulties, there are still some to be surmounted, but as occurred in the development of the large gas engine, experience has enabled defects to be remedied and extreme reliability in working to be possible. Combustion in the Diesel engine is a most important feature, and whilst the general laws are known in this connection, there is an immense field in actual working practice of which little is known. As progress is made it will become possible to build larger units, but whether the type will be two or four-cycle, or double acting, or of the opposed piston class is not at the moment easy to forecast. In my own opinion when the cylinder diameters commence to exceed 45 ins. to 50 ins., then a critical stage will have been reached and considerable research will be necessary in many directions. At the present time our cylinders are roughly in the neighbourhood of 30 ins. as a limit. I am very much interested in the old question of air compressors and some difficulties in operation, such as rusting in the intercoolers, purgepots, errosion, etc. It was recently advocated that certain internal iron or metal exposed parts be treated with a hard drying graphite compound which arrested rusting. Whilst, however, mineral acids have been responsible for corrosion and erosion, it must not be assumed that the lubricating oil is in all cases the root cause of the trouble, as frequently the corroding agent may be traced to condensed moisture and high temperature compressed air, which form the base of the corrosive element. Close attention therefore to the draining of water from the compressor system at regular intervals is of first importance. To minimise the danger of explosions in the intercoolers, due to high temperature and excess lubricating oil, it should be clearly understood that under no circumstances should the L.P. compressor be allowed to have air intake restricted, as any wiredrawing at this point causes a vacuum which the oil may reach, and on this happening, pulverisation occurs and the mingled compressed air and oil vapour in its passage further in the compressor system, reaches a temperature stage when conditions are favourable to an explosion. It is desirable, therefore, where the compressors deliver a volume in excess of the blast, bottle or receiver requirements, either to leave the drains slightly open on the compressors or arrange that air unrequired is blown off at the relief valves provided.

A point of much importance in Diesel engine operation is to observe closely unequal temperatures at the various points of the water cooling system, as recorded by the temperature gauges, as after a voyage or so on reference to the daily log a reliable mean will have been recorded, and any wide variations, whether from the water-cooled exhaust valves or from the pistons, from the main barrel of the jacket, or from the breech end, will need to be closely scrutinised so that the reason for any departure from uniform working temperatures may be ascertained.

NOTE.

Subsequently a member pointed out that in the paper, on page 262, it is stated that the exhaust valve opens 40° before B.D.C. and closes 10° after T.D.C. Inlet valve opens 20° before T.D.C. and closes 15° after B.D.C., and that if this were so, the closing of the exhaust valve would overlap the opening of the inlet valve by 30° and lead to very heavy smoking. The more probable reading is that the inlet valve opens 20° after the top dead centre, which gives both valves closed for 10° before inlet valve opens, this eliminates the possibility of smoking from this source.

REPLY BY THE AUTHOR, HOMER McCRIRICK.

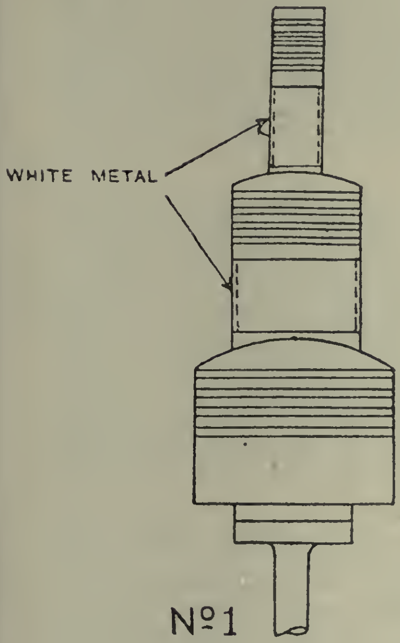
In reply to the discussion which followed the reading by Mr. Adamson, of my paper on "The Maintenance of Marine Diesel Engine Installations," a number of points were raised during the discussion which I think can stand for a few further remarks. A great deal of interest seems to have settled round my comments regarding safety valves on compressor intercoolers, and the damage that can be caused by their failure to act when required.

NOTES ON THE MANAGEMENT OF MARINE DIESEL ENGINE INSTALLATIONS.

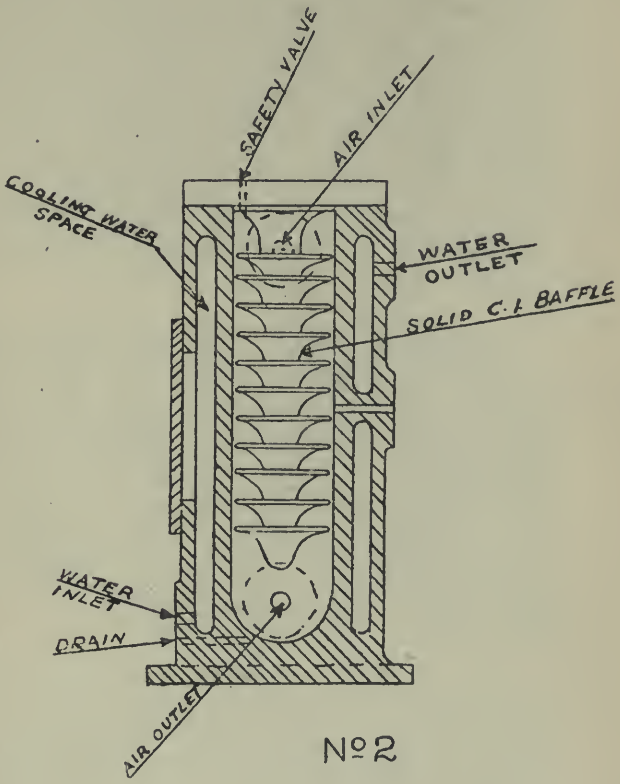
I had the experience some time ago of an explosion taking place inside the L.P. air casing of a two-stage air compressor, happily I may say without any disastrous effects. The engine was a four-stroke Diesel, with a two-stage air compressor direct driven from the end of the crank shaft for supplying blast air. I had just started up the engine when there was a loud report, a flash and then smoke, and on examining the air compressor, the safety valve on the L.P. air casing was found to be leaking continuously. The engine was stopped, and the valve taken out for examination, when it was found that the spring had lost its compression in the first three coils and from their colour they appeared to have been greatly overheated by the passage of hot gases over them.

The conclusion arrived at was that through an accumulation of oil in this casing, an explosion had occurred; whether this oil was a lower flash-point variety than should have been found there, could not be ascertained. The intensity of the explosion could not be judged, but from the heat imparted by the gases when passing the spring, it must have been considerable, and had the safety valve not been in a condition to release the force of the explosion, considerable damage might have been caused to the casing.

Whilst on the subject of safety valves, I think it will be interesting if I relate an accident that came to my notice a short time ago. The vessel, which has been in service since 1911 is one of the pioneers of the present-day motor ship, and, so far, has had a very successful career. The machinery consists of a four-stroke directly reversible Diesel engine with a three-stage air compressor direct from the main shaft. The piston for this three-stage air compressor is fitted with white metal, as shown in sketch No. 1, for the purpose of reducing friction between the walls and piston. During this particular voyage, overheating of the air compressor piston occurred to such an extent that the white metal became partially melted. The engine was therefore stopped and the piston taken out and replaced with the spare one. When things were all coupled up, the engine was again started and everything seemed to be in order. In less than an hour a loud report was heard in the engine-room and on examination being made it was found that the I.P. intercooler had burst and scattered in all directions; water service and oil service pipes in the vicinity of the same were crushed and fractured, and a bulkhead three feet away had a hole pierced in it three inches long by two inches. This particular intercooler was made of cast iron, and a section of the same is shown in sketch No. 2. Various parts of the cast iron were examined for deterioration that would have been likely to



Nº1



Nº2

account for the bursting, but no extensive wasting could be found and therefore the cause of the accident had to be looked for in other places. The outlet pipe from the cooler, although badly fractured and twisted, appeared to have been choked, so it is to be assumed that some of the white metal and carbonised oil from the piston found their way into the cooler, and after the engine had run for a little while the thin pieces of white metal must have become fixed in such a position as to choke the discharge pipe.

When this stoppage occurred, the pressure started to rise above normal, and as the safety valve failed to lift, it resulted in the cooler bursting. Had a thin disc been fitted in some place on this cooler the bursting would have been avoided; on the other hand also, had the safety valve been in working order the engineer would have had warning of an excessive rise of pressure and taken steps to remedy the same. Take the case of the explosion now, this I consider much more dangerous than a gradual rise of pressure, as the safety valve was able to release the sudden increase of pressure. Of course there is no reason why we should not have as many safety devices as possible, but I think if an engineer knows that all his safety valves are in working order he can go about with very little to worry him as far as bursting of air receptacles goes. Where I do think that pressure discs could be fitted to an advantage are at various places along the air starting and blast pipe lines, as should fuel oil find its way into them, there are chances of a violent explosion taking place.

A member raises the point about deterioration taking place on the steel shells of air vessels. I had the opportunity of examining internally the two starting air vessels on the ship I have just mentioned in connection with the bursting of the intercooler. These vessels carry air at 15 atmospheres, and considering they have been in use for almost 10 years they are in excellent condition, the only place where I found slight pitting was round the drain, and in no case were the recesses more than 1/16th inch deep.

As regards blast bottles, I have examined a few of them and in all cases the working bottles had a coat of oil adhering to the walls. The usual practice for cleaning these bottles is to put in soda and boil out the bottle with steam. I am not in a position to say to what extent pitting has been found in these small bottles as they are very difficult to examine at the bottom, and comparisons cannot readily be made between the working blast bottle having a coat of oil and the spare blast bottle with none.

The wastage of lubricating oil taken at about 1/14th of a ton per day for an installation of 3,000 I.H.P. is, I think, quite a fair figure, and there is no doubt that this will be improved upon as the sizes of engines increase and number of working parts decrease. Take the case of cylinder lubrication; in the older and smaller ships in which the pistons were not so efficiently cooled, the consumption of cylinder oil would average six gallons per 1,000 I.H.P. per day, whilst in the latest ships the cylinder oil consumption averages $1\frac{1}{4}$ gallons per 1,000 I.H.P. per day, which is quite a substantial reduction.

In reply to remarks made concerning the average temperature during the power stroke, it is assumed that the engine was working at a mean pressure of 95 lbs. per square inch. Varied opinion still exists as to the true temperature conditions found during the burning of the fuel in the power stroke of a Diesel engine, and as far as I am aware no general formulæ seems to be agreed upon for this. The temperature of 2,200° Fah. was obtained from some experiments carried out on the Continent. In Fig. No. 2 in the paper, the temperature at C is stated to be about 2,200° Fah. Under the paragraph of "Cylinder Heads" the 2,200° Fah. is intended to be the average temperature during the burning of the fuel in the power stroke.

With reference to further remarks regarding keeping record of the weights of the H.P. compressor coils so that any thinning of the same may be noticed and the coil condemned before it bursts. I heard of this coil weighing proposal some time ago, but seemed rather dubious of it. A certain quantity of compressor lubricating oil is bound to find its way into the coil and adhere firmly to the inner walls, so any record of weights may be misleading. I have not seen any H.P. coils opened up, but judging from the quantity of oil found in the tubes of the I.P. and L.P. intercoolers the weight of oil in the H.P. cooler coil must be considerable.

One suggestion I have heard, which would enable engineers to know when a coil has become worn to any extent by the scouring effect of the air, is to sink the point of a drill a certain depth at various places along the coil. When the metal remaining between the drill point and the inside of the coil is worn away, a small air leak will commence and it may then be assumed that the coil has worn a thickness equal to the difference between the original thickness of metal and the depth to which the drill point has been sunk. This member also points out the inadvisability of shutting off too many cylinders; I have mentioned the cutting out of cylinders in the paper, but it was not intended that when a valve in a cylinder cover gives out that the fuel and blast air are to be shut off and nothing more thought of it until port is reached.

Take the case of a fuel valve giving out; if it is very far gone the fuel and blast air must be shut off from the same immediately, or the engine stopped. The damaged valve can be taken out and replaced by a spare valve in half-an-hour, but it may be more convenient to keep the engine running on five cylinders for a short time until the spare valve has been examined and made ready.

The same holds good with a cracked piston; if the fuel and blast air are not shut off soon after it is discovered that the piston is not being circulated by cooling water, there are chances of the piston overheating and perhaps seizing in the cylinder liner.

The replacing of a piston may mean six hours work and there is no reason why the engine cannot carry on under reduced power until the spare piston has been cleaned and put in order ready for dropping in place.

Most engineers realise the extra strain set up in crank shafts through cylinders being cut out, and I do not think any of them would carry on with more than one cylinder out of action if a repair could be effected. Even with engines of the most reliable design and running under experienced engineers, there is still the possibility of a cylinder cover valve giving out, and I am sure allowance to take up the extra stresses caused by this on the crank shaft is made by the designers.

From the above remarks on the changing of cylinder cover valves and pistons, I hope members will not carry away the false impression that the failure of valves or pistons is a regular occurrence. Most of the larger motor ships are now covering distances from 4,000 to 8,000 miles without any involuntary stoppage, the valves being overhauled at regular intervals when the vessels are in port. Also when care is exercised in the lubrication of the pistons, they are able to remain in operation for distances from 60,000 to 70,000 miles without having to be drawn for cleaning the rings.

Cracked cylinder covers are certainly a source of worry, but so far, I have not heard of any serious breakdowns due to an accumulation of water in the cylinder of a large engine, where the compression clearance is usually about two inches.

If a cover is badly cracked it must be changed, but if the crack is small it is possible to avoid any large quantity of water finding its way into the cylinder when the engine is being manœuvred.

I am not in a position to say anything with regard to the opposed piston type of engine, but I can only endorse the remarks on the position of the Diesel engine being unassailable, and though the four-stroke is considered the reliable engine at present, I do not think the time is far distant when the two-stroke engine will come to the fore as being the more suitable engine for single screw vessels from 250 to 350ft. long.

In reply to a subsequent note from a member as to the timing of the exhaust and air inlet valves as stated in my paper.

The valve timing stated in the paper is as intended, the exhaust valve closing 10° after the top dead centre, and the inlet valve opening 20° before the top dead centre, this gives the valves an overlap opening of 30°.

The object of making the valves overlap in their opening is to expel as much of the burnt gases in the clearance space as possible. The presence of burnt gases in the cylinder during combustion is detrimental to the proper burning of the fuel and may tend to cause smoking.

Consider the exhaust stroke of the cycle with the piston moving upwards and the gases flowing through the exhaust valve until such a time as the piston has almost reached the top of its stroke. It is now assumed that the gases in the exhaust pipe have attained a certain velocity moving in a direction away from the exhaust valve and the tendency for this moving mass will be to create a vacuum behind it and draw more gas from the cylinder.

The inlet valve having opened 20° before the top dead centre the air will take advantage of the partial vacuum created in the cylinder and have time to acquire a certain velocity by the time the piston has reached the top dead centre. This air flowing into the cylinder when the exhaust valve is still open will have a scavenging effect in the cylinder, and when the exhaust valve does close, the clearance volume of the cylinder should have a large proportion of pure air within its walls, which it otherwise would not have had but for the overlap of the two valves.

SESSION 1920-21.

President: LORD WEIR OF EASTWOOD.

Reversing of Marine Oil Engines.

BY WALTER POLLOCK (Member).

Read Tuesday, January 4, 1921.

IN this paper a few of the methods adopted for reversing marine oil engines will be described, all technical details of construction and criticism of design being avoided. The author has been called upon to prepare this paper at very short notice, otherwise a larger number of makes would have been included and large scale drawings to more clearly illustrate them, would have been supplied.

Every propeller should be capable of being reversed in order to stop the vessel and go astern. To be efficient, the propeller must reverse rapidly and the method must be very reliable.

In ordinary marine steam engines, the Stephenson's link motion and valve gear has been so perfected, that it can be termed absolutely reliable. With turbine machinery the problem is somewhat difficult, and with oil engines it is very difficult.

The systems in use for oil engines, when the varying details are taken into consideration, number several hundred, but they may be classified as follows:—

Reversible blade propellers.
Reverse gears.
Direct reverse by pre-ignition.
Direct reverse by air.

Propellers with reversible blades are largely used in fishing vessels, and have the advantage of flexibility with the engine always running one way, and at more or less constant speed. About 300 H.P. appears to be the limit of size for this type, which requires hollow stern shafting or a sleeve arrangement to operate the blades, usually in connection with racks and pinions. It is, however, unusual to find this arrangement on engines of over 65 B.H.P. It will be seen that with even slight wear, which must occur sooner or later, the variation in the angle of the blades must be considerable. The blades can, however, in case of damage be renewed economically, though on the other hand damage to the propeller may cause costly renewals to the rack, etc. These propellers require an unduly large boss in which to house the mechanism for reversing.

Large numbers of fishing vessels have been fitted with propellers with reversible blades, because of the ease with which the speed of the vessel can be controlled whilst fishing with the lines out or the nets down.

It is claimed that small engines run with greater smoothness with this type of propeller, and since the fishermen like the arrangement, the question of economy is not considered by them. A few ferry boats and other craft are also fitted with this method of reversing, but its scope is very limited.

Ships' launches, naval motor boats, river and harbour launches and small barges having petrol and paraffin engines are fitted with reversing mechanism of the toothed gear variety either of planetary, bevel or silent pitch-chain type.

The first kind is seen in Fig. 1, which shows the Langdon reverse gear.

The action is as follows:—

On the after-end of the shaft (A) is keyed the clutch housing (B) which carries the clutch plates (M). The interlocking clutch plates (L) which are free to slide fore and aft are carried on the drum (K) which is keyed to the clutch shaft (J) and are pressed into engagement with the plates (M) by the spring (N). When the plates are once engaged the pressure of the spring is reinforced by the action of the rollers (R) which are jammed by the rotary motion of the shaft between the inclined plates (P_1) and (P_2) on the two clutch members respectively as shown in the upper illustration of the Fig. 1. Thus the plates are locked practically solid and there is no need to fear a slipping clutch.

The sun and planet system is carried in the reverse gear casing (D) which is free to revolve on the bush (C) in the clutch housing and through the bush (E) on the sun wheel (F) which is keyed to the thrustshaft. The remainder of the system consists of the planet pinions (G) and (H) in the reverse gear casing and the pinion (I) driven by a feather in the clutch casing (B).

The gear as shown is in the position for ahead running, the clutch plates being pressed together by the spring (N) and by the inclined planes so that the clutch housing drives the reverse gear casing through the pinion (I) and the whole gear revolves as a single unit.

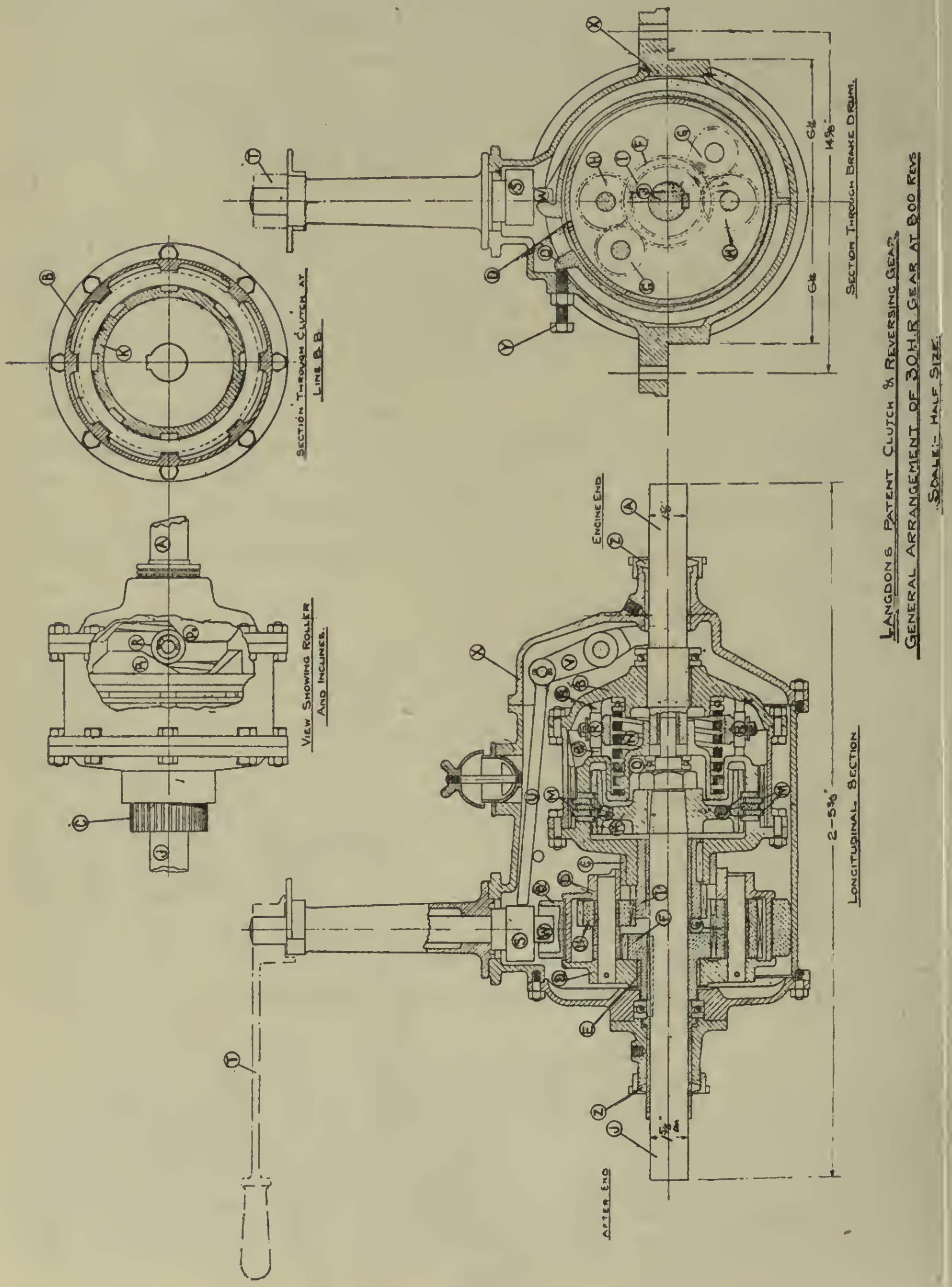

Langdons Patent Clutch & Reversing Gear.
General Arrangement of 30 H.P. Gear at 800 Revs
Scale:- Half Size.
Section through Brake Drum.
Section through Clutch at Line B.B.
View Showing Roller and Inclines.
Longitudinal Section
Engine End
After End
6¼
14⅝"
2-5⅜"
1⅝"
A
B
C
D
E
F
G
H
I
J
K
L
M
N
O
P
Q
R
S
T
U
V
W
X
Y
Z

Fig. 1.

When the reversing lever (T) is turned through about 45 degrees the cam (S) moves the rod (U) and with it the forked lever (V) which presses against the ball race on the shaft (A) and so forces the shaft and the clutch casing (B) aft, thus disengaging the plates (L) (M) since the clutch drum (K) is on a separate shaft (J), which is prevented from movement fore and aft by the thrust block (not shown). With the lever (T) in this second position, the pinion (I) is driving the planet pinions (H) and (G) idly round the "sun" wheel (F) but not rotating either it or the thrust shaft (J) owing to the clutch being disengaged. The engine is therefore running free.

To go astern, the lever (T) is turned athwartships which rotates the cam (W) against a lug on the brake band (Q) which therefore holds the reverse gear casing (D) and prevents it from revolving, the clutch plates still remaining disengaged owing to the action of the cam (S). The planet pinions (H) and (G) can now no longer revolve idly round the sun wheel (F) but owing to the rotating motion imparted to them by the pinion (I), they drive the sun wheel, but in the opposite direction to that of the shaft (A), and consequently the boat is driven astern.

The whole gear is enclosed in an oil-tight casing (X) with glands (Z) (Z) to prevent the oil escaping.

Provision against wear of the brake band is made by fitting an external setscrew (Y) which can very easily be screwed up half a turn or so whenever there is any sign of slipping or tendency not to go astern.

Fig. 2 shows the "Ideal" reversing gear which was employed by Messrs. Beardmore on their surface ignition engines which were not directly reversible.

Here the reverse gear casing (A) as before containing the sun wheel (C) keyed to the engine shaft and the sun wheel (B) keyed to the thrustshaft and the planet pinions (not shown) which engage therewith. The male clutch cone (L) is again keyed to the thrustshaft, but there is no brake band as in the Langdon and its place is taken by the female clutch cone (N) which is bolted to the framework. To go ahead, the collar (F) is caused to slide aft along the engine shaft (G) by the reversing lever (not shown) which pushes the rollers (H) along the inclined planes on the toggle pieces (J), which are thereby rotated about the centres (K). This causes the spindles (E), which are borne in the reverse gear casing and screwed into the male and female floating cone rings (D), to move forward and engage the

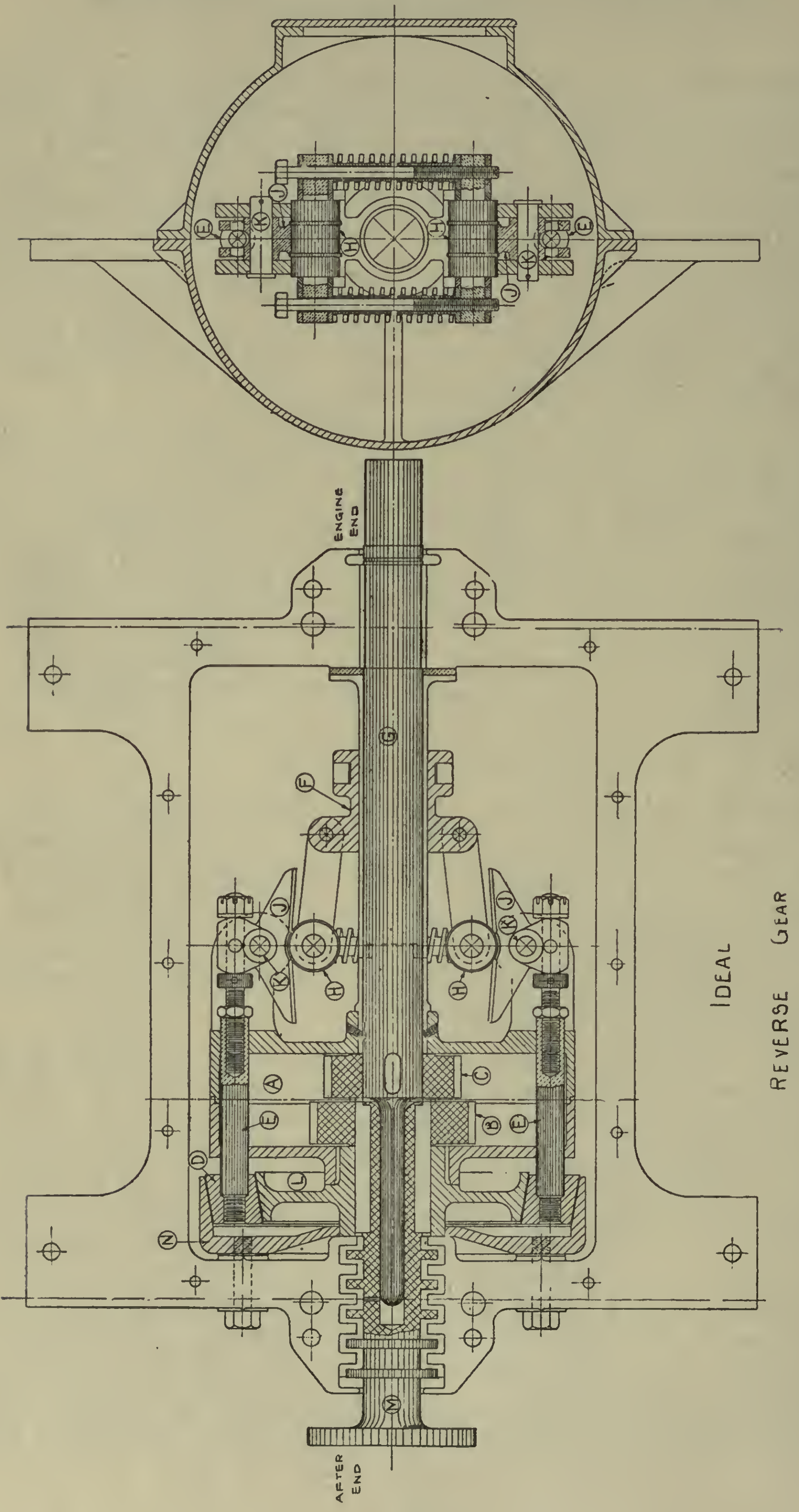
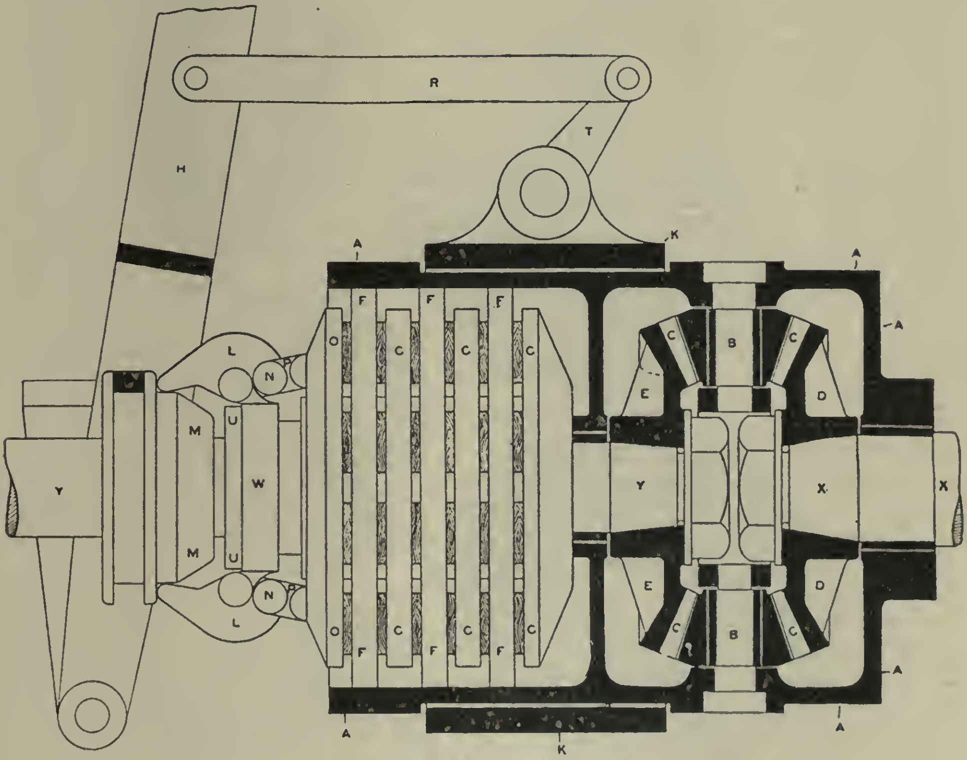

E
K
J
H
H
J
K
E
ENGINE END
G
F
J
K
H
H
K
J
A
C
E
B
E
D
L
N
M
AFTER END
IDEAL REVERSE GEAR

Fig. 2.

female cone (D) with the male cone (L) and so drive the thrust-shaft ahead. In the neutral position as shown, the rollers (H) are in the centre of the face of the toggle pieces and the floating ring (D) is clear both of (L) and (N). To go astern, F slides in a forward direction and the rollers (H) mount the other inclined planes of the toggle pieces and the floating ring (D) is then pushed aft into engagement with the fixed clutch ring (N), so that the reverse gear casing is held from revolving, as with the brake band of the Langdon gear. The sun wheel (C) will then drive the sun wheel (B) in the reverse direction through the planet pinions in the reverse gear casing, and the boat will go astern.

A well known type of bevel reversing gear is the Kromhout, (Fig. 3), the various parts of which are shown dismantled in



REVERSE GEAR.

Fig. 3.

Fig. 4. Referring to the sectional drawing, the reverse gear consists primarily of a cast-iron casing (A) which is mounted

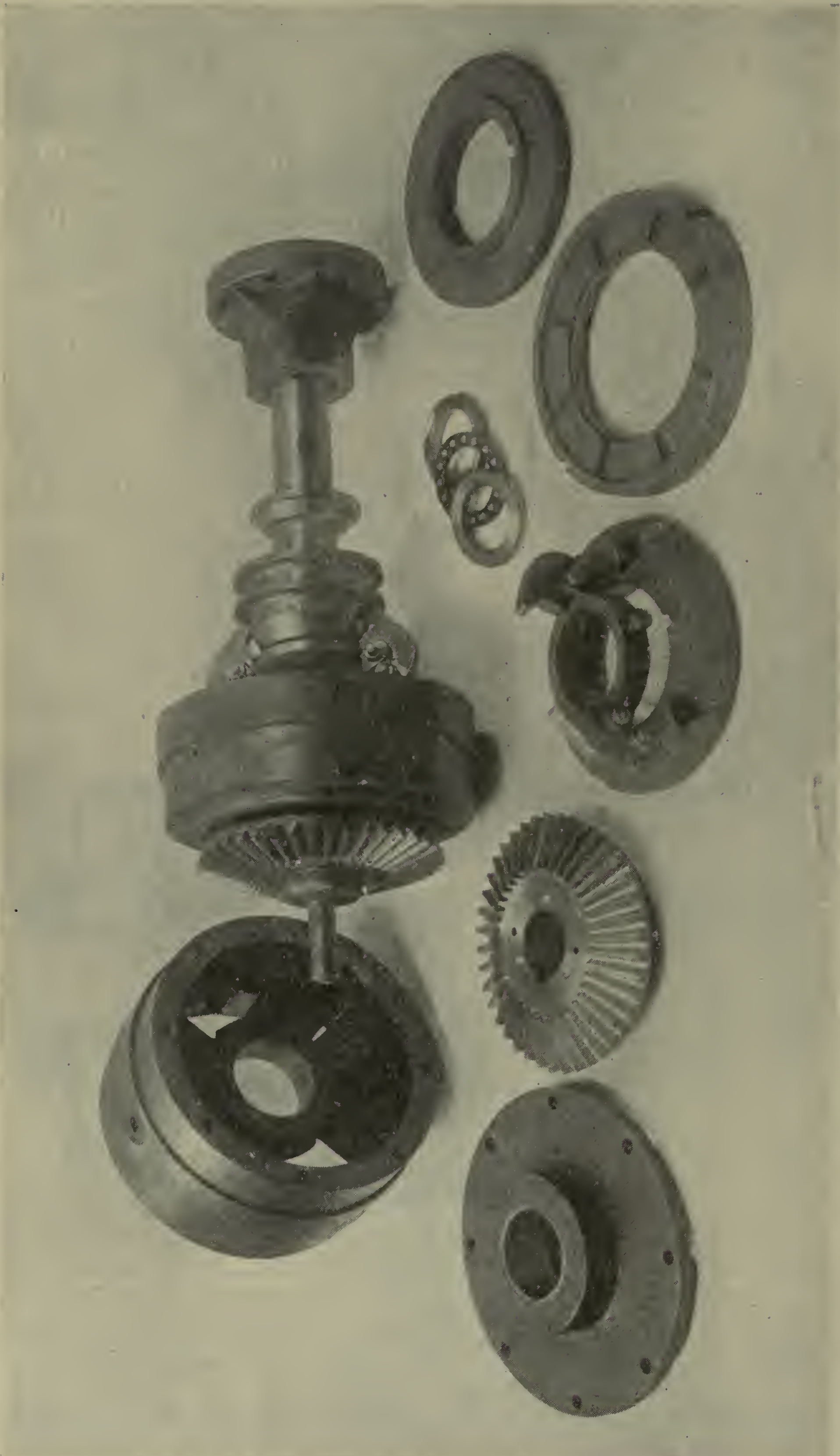


Fig. 4.—Details of Reverse Gear.

loosely on the engine shaft and the thrustshaft (which are quite separate from each other) and which carried on its forward end spindles BB on which the bevel pinions CC are free to revolve. Engaging with these bevel pinions are the bevel wheel (D) keyed to the engine shaft (X), and the bevel wheel (E) keyed to the thrustshaft (Y). On the other end of the case are fitted friction clutch plates FFF, which are capable of a small movement fore and aft, but which are prevented from revolving relatively to the case by means of long keys in the case. Carried on the propeller shaft are corresponding plates GGG, which also have free movement fore and aft, but no rotary movement relative to the shaft.

The operating gear consists of the reversing lever (H), the brake band (K), and the clutch actuating levers LL. The action of the gear is as follows:—

The neutral position is obtained by placing the reversing lever (H) in the central position, where the clutch plates FFF GGG will be disengaged and the brake band (K) will be free; the bevel wheel (D) will be revolving with the engine shaft, while the thrustshaft (Y) and bevel wheel (E) will tend to be kept stationary by the resistance of the propeller. The bevel wheel (D) will therefore "wind" the pinions CC round the stationary wheel (E), and cause the case (A) to revolve idly through the spindles BB attached thereto.

Now if ahead gear is wanted, the reversing lever (H) is pushed forward and the hardened nose of the clutch actuating levers LL mounts the hardened cone (M), which causes the opposite end (N) of the levers LL to exert a powerful thrust on the disc (O) through the links PP. This thrust causes the plates FFF and GGG to interlock by friction, so that the case, which is still revolving as before, now carries round the propeller shaft with it through the clutch plates, while the pinions CC and the two bevel wheels (D) and (E) revolve as a solid mass without any motion relatively to each other. There are thus no gear wheels in action for ahead gear, but a direct drive through a powerful friction clutch.

For astern gear the reversing lever is pulled back, which disengages the clutch plates and at the same time by means of the link (R) pulls over the little lever (T) the boss of which has its faces cut at an angle so as to form inclined planes. These inclined planes cause the brake band (K), which is supported on the bedplate concentrically with the shaft, to grip the case and prevent it revolving. Thus the pinions CC can no longer travel round with the wheel (D) and therefore impart motion to the wheel (E) in the direction opposite to that of the wheel (D), and so a reverse motion is transmitted to the thrust and propeller shaft.

Reversing gears of various types have also been adopted in a large number of surface ignition oil engines, and the limit of size in this case would appear to be 250 B.H.P.

The advantage of fitting a reversing gear is that the engine can be of the simplest design and construction, and need only run in one direction both for ahead and astern motion of the ship. The disadvantage is the introduction of gear wheels, either of the "sun and planet" or the ordinary bevel gear type. All designers recognise the difficulty of designing a gear that will be light and that will nevertheless withstand the sudden torque to which it may be submitted from the action of the engine itself or from a blow due to an obstruction striking the propeller, while the question of wear has always to be considered.

To reduce the severe strains on the gear during reversing, it is necessary to stop the tail shaft and propeller from revolving before the reverse gear is actually engaged; various methods to attain this end will be referred to later.

On the Continent, several simple methods of reversing may be seen, that do away with gear boxes, such as a clutch fitted to the main drive and a countershaft with pulleys and a crossed belt to reverse the motion, a design that is only used for river and canal work.

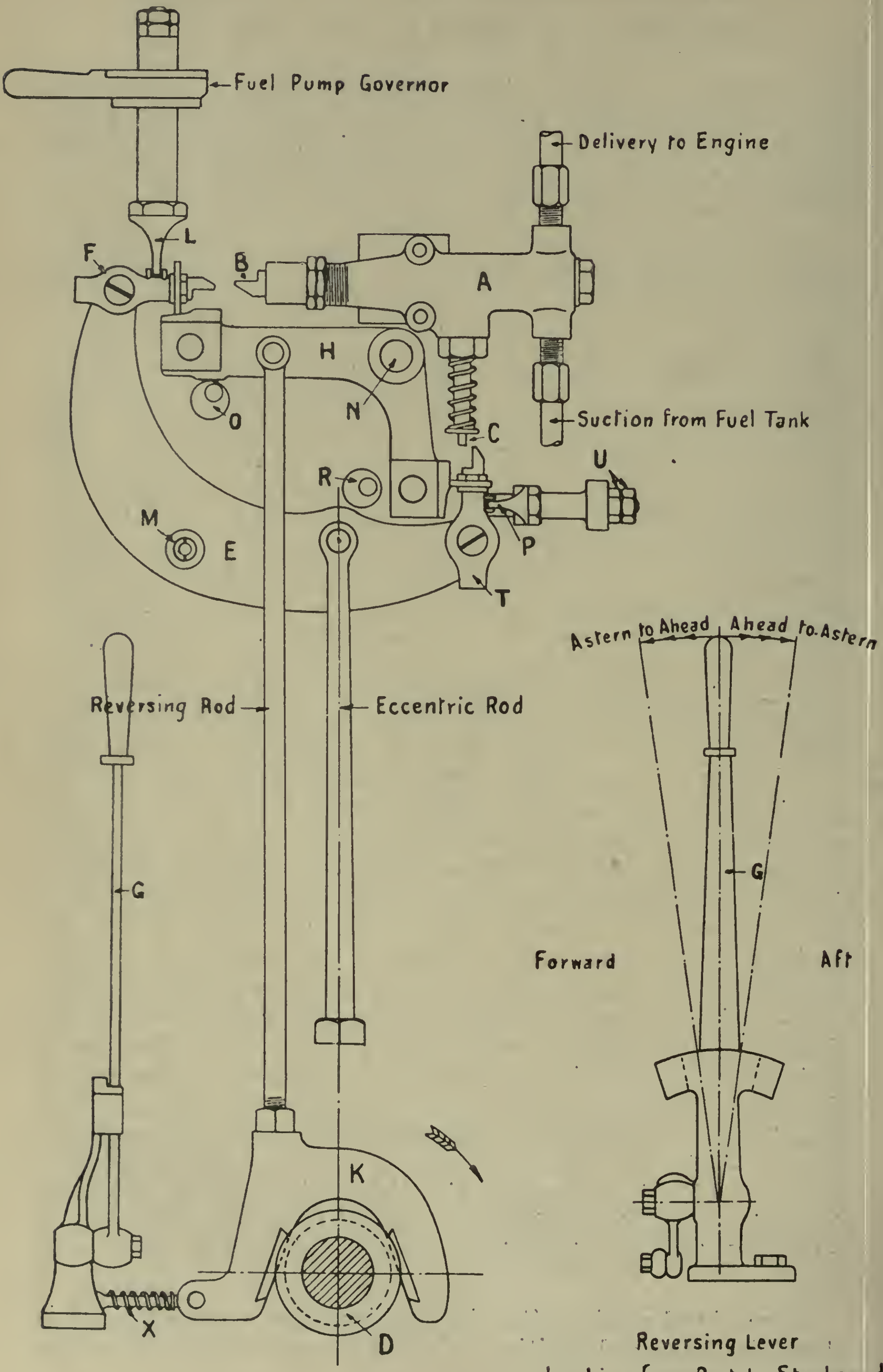
For an oil-engined tug used on inland canals, the author fitted a propeller at each end of the vessel, one right-handed and the other left-handed, with a clutch at each end of the engine. To reverse the direction of motion of the tug it was only necessary to move the clutches over and bring the propeller at the other end of the vessel into action. The remaining shaft and propeller being de-clutched were free to revolve so as to minimise the drag or resistance.

Before dealing with the more important methods of reversing, mention might be made of the Kitchen Rudder system, which has been designed to avoid the necessity for the engine or propeller to reverse at all. By the closing of the two parts of the rudder the water thrown aft from the propeller into the rudder cavity is deflected forward and so causes the reversal of the direction of motion of the vessel. This has apparently proved very effective for small craft, and is gradually being introduced into large vessels. There are many other advantages claimed for this patent rudder, which do not particularly concern the question of reversing the motion of the ship.

Direct reverse by pre-ignition is adopted in a few well known makes. Probably the best known of these is the Bolinder, of which the patented direct reversing arrangement is shown in Fig. 6.

Here A. is the fuel pump which has two plungers, B. the main and C. for reversing. When going ahead the arm E. is rocked by the eccentric rod which causes the tappet F. to ride up the little inclined plane shown at the end of the bell crank H. and comes into contact with the the fuel pump plunger B. and so drives a charge of fuel into the cylinder. If the engine speed becomes excessive, the momentum imparted to the tappet by mounting the inclined plane will be too much for the flat spring L. and throws the tappet clear of the pump plunger, and a stroke will be missed and the speed reduced. In order to go astern, the clutch must first be disengaged and the reversing lever G. then moved into the astern position, which, through the operation of the bell crank and link shown in the perspective view will press the "U"-shaped piece K. against the grooved disc D. mounted on crankshaft. The friction between K. and D. will lift the horizontal arm of the bell crank H. and so lift the tappet F. out of reach of the plunger B. and the engine will commence to slow down. When the predetermined speed decided upon for safety has been reached the flat spring P. will then overcome the inertia imparted to the tappet T. by the inclined plane on the vertical arm of the bell crank H. and allow the tappet to come into contact with the reversing plunger C. and inject a charge of fuel into the cylinder just before the piston reaches the top centre which drives the piston back again and the direction of rotation is reversed. The forked piece K. is then lowered due to reversal of shaft and disc D. and so bell crank H. resumes its normal position allowing tappet F. to move main fuel pump plunger B. and so the engine continues running in the astern direction and receiving fuel from the main pump B. It is to be noted that pump plunger C. is only used for reversing either from ahead to astern, or *vice versa.*

To go ahead again, after declutching, the reversing lever is placed in the ahead position, when the same sequence of opera-



Transverse View Looking Forward.

Reversing Lever
Looking from Port to Starboard.

Fig. 6.

tions takes place, the whole operation being automatic when once the lever is set for the desired direction of rotation.

Flywheels appear to be necessary in all types of internal combustion engines with any number of cylinders up to four or six, or even more, although it is difficult to see why engines with as many as eight cylinders should require this heavy and cumbersome addition. With only two or four cylinders, it is necessary for smooth running, although it must add to the difficulty of reversing, as the flywheel must be brought to rest before reversal can take place.

Again, most of the engines with pre-ignition reverse and reverse gears have large and awkward clutches to disconnect the stern shafting before reversal takes place, whereas, in a steam engine, the stern shafting and propeller which take the place of a flywheel are considered to be sufficient for the smooth and even running.

Whilst on this subject, it is interesting to note that some engineers object to the pre-ignition system of reverse on the ground that the crankshaft has to resist an excessive torque, seeing that the engine has to absorb the momentum of a massive flywheel and start it rotating in the opposite direction. Surely the stresses on the crankshaft are not greater than in a steam job where the stern shafting and propeller are directly coupled. It would therefore appear that the ideal engine would dispense with the flywheel and also with the clutch, even for small powers, which would be a great advantage, as it would reduce the length and weight of the engine, and increase the rigidity of the bedplate.

Even with compressed air reversing systems, the disadvantage are almost as great.

To reverse an oil engine by air, it is of course necessary to have an air compressor, and where only one of the cylinders is used for reversing, as in the case of the Plenty engine with air (and the Bolinder with pre-ignition), a clutch between the engine and propeller also. The limit of size for this method appears to be at present 500 B.H.P. per unit.

For oil engines of large power, the general practice is to use starting air valves on all cylinders, thus dispensing with the necessity of the clutch which at all times must be a disadvantage.

In the case of small engines, an air compressor may not appear to be a serious item, yet if the prices of several makers are studied it will be recognised that this item considerably increases the cost of the engine.

Where camshafts, tappet rods, valves and gear are used, the cost of manufacture and the weight of the engine are increased, and a number of additional parts are subject to wear.

Where an air compressor is required for atomising and mixing the fuel oil, the most economical method of driving it is undoubtedly to couple it direct to the main engine, but the disadvantage is that it increases the length of the main machinery. In any case, whether the air compressor is driven from the main engine, or as a separate unit, it requires a duplicate set as a stand-by.

For reversing only, a small air compressor with large air reservoirs will best answer the purpose, because a quantity of air is ready stored for immediate use.

Naturally an engine which is not fitted with a clutch but is connected solidly to the propeller, requires a much larger compressor and compressed air reservoirs.

When the other auxiliaries necessary for ocean going ships propelled by internal combustion engines are taken into consideration, the large number of engines and auxiliaries seen in the engine-room is not surprising, although somewhat unusual to the ordinary steam engine engineer.

An example of direct reversing by compressed air in conjunction with a clutch is that fitted by Messrs. Plenty, and illustrated in Fig. 7. The ordinary camshaft is not used, but instead the vertical governor shaft driven from the crankshaft at the after-end of the engine has two cams, one for ahead and the other for astern, mounted on a sliding sleeve. These cams are set to open valves for the admission—to the proper cylinder—of compressed air just after the piston has passed the top dead centre in either the ahead or astern direction of rotation.

In this system, the operation of the reversing wheel alone is sufficient for cutting off the fuel injection, admitting air for reversing the engines, starting the fuel injection again for the reverse direction, and finally shutting off the compressed air. This is carried out by means of a hand-operated camshaft through bevel gears when the manœuvring or reversing wheel is turned.

The action is as follows:—

Referring to Fig. 7, the vertical shaft (4) is driven by spiral gearing off the after-end of the crankshaft, and is geared to run at the same speed as the crankshaft. The cams (Q) and (R) are machined solid with the sleeve (13), which is free to move vertically on the shaft (4); the movement of this sleeve is controlled by the lever (20) from the scroll cam (T) on the shaft (X), which is rotated through bevel gears by the manœuvring wheel (16), Of the cams (Q) (R), one is for ahead and the other for astern,

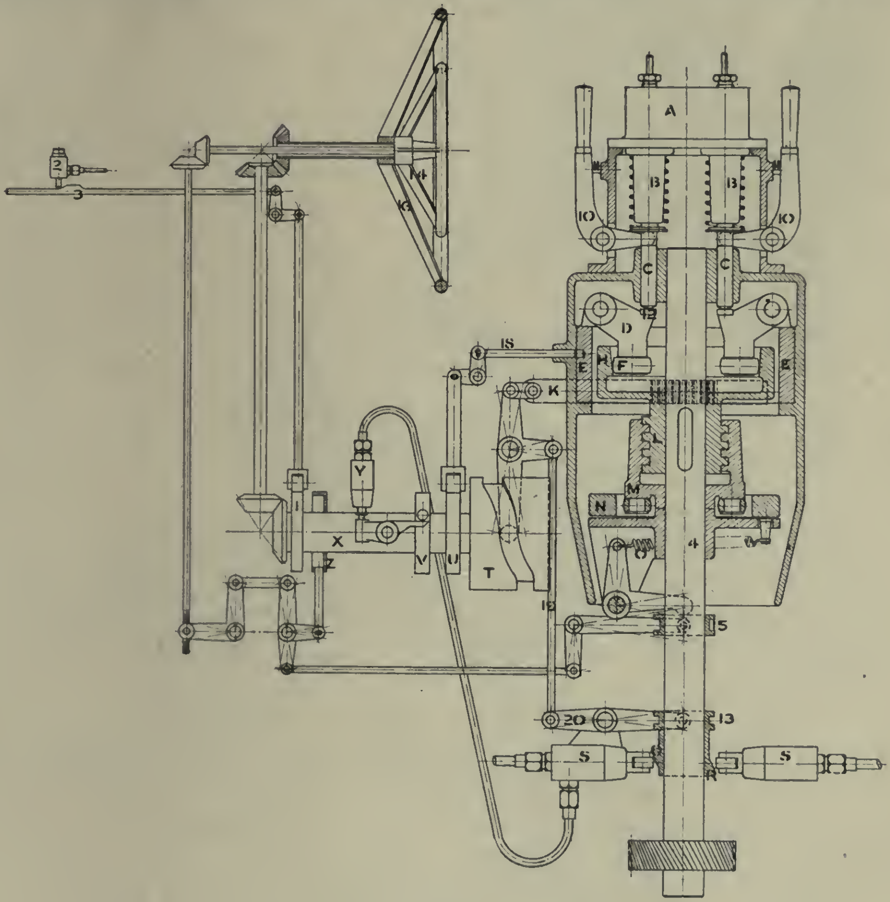


Fig. 7.

and according to the direction in which the wheel (16) is turned, so the ahead or astern cam opens either of the valves (SS) and so admits air to the proper cylinder for turning the engine ahead or astern.

The first effect of turning the handwheel is, however, to cut off the fuel injection by opening the bye-pass (2), the rotation of the shaft (X) causing the nose of the cam (I) through the medium of the links shown, to slide the cam (3) under the plunger of the bye-pass valve. Practically simultaneously, the cam (V) on the shaft (X) opens the master air valve (Y).

At the same time, the scroll cam (T) is causing the air to be admitted to the valves (SS) for either ahead or astern through the lever (20) and through the rack (K) is rotating the sleeve (E) to which the fuel pump levers (D) are connected, and so altering the timing of the fuel injection to suit the ahead or astern direction of rotation. The cam (U) fixed to the shaft (X) by means of the plunger (18) locks the sleeve (E) when once placed in position for the required direction of rotation. In the position shown in the diagram no air is being admitted, and owing to the position of the cam (I), the fuel bye-pass valve (2) is shut, and therefore the engine is in a position for running either ahead or astern on fuel.

The wheel (14) is not in any way connected with the reversing mechanism, but only controls the speed by sliding the collar (5) and so altering the tension of the governor spring (0), which varies the stroke of the fuel pump plungers. The hand levers (10) are only for priming purposes in the usual manner.

A number of surface ignition engines of the higher powers use compressed air reversing systems carried out by means of camshaft operated valves, a separate valve and two cams (one for ahead and one for astern) being used for each cylinder. An example is that shown in Fig. 8, which shows a type used on Kromhout marine oil engines. The camshaft which can be seen at the lower part of the cylinders is driven by gearing off the forward end of the crankshaft. Two cams are fitted in the way of each cylinder for operating the air valves, one cam only, of course, being in operation at a time, while four additional cams are fitted for operating the fuel pumps (A). The camshaft is moved in a fore and aft direction by means of the hand wheel (B), and can be moved either with the engine stationary or when running, the cams being arranged to admit compressed air to that cylinder which is in the best position for starting the rotation of the engine in the required direction, either ahead or astern. The operation of the wheel (B) alone is sufficient to start or reverse without having to perform any other operation.

The air valves (C) are seen at the tops of the cylinders, each one connected to the compressed air pipe. The hand lever (E) is only for altering the stroke of the fuel pumps and so varying the speed of the engine.

The hand wheel (G) operates the friction clutch for disconnecting the propeller shaft from the engine, but this is only used when it is necessary to stop the vessel.

In order to go from ahead to astern to take way off the ship, it is only necessary to turn the hand wheel (B) in its proper

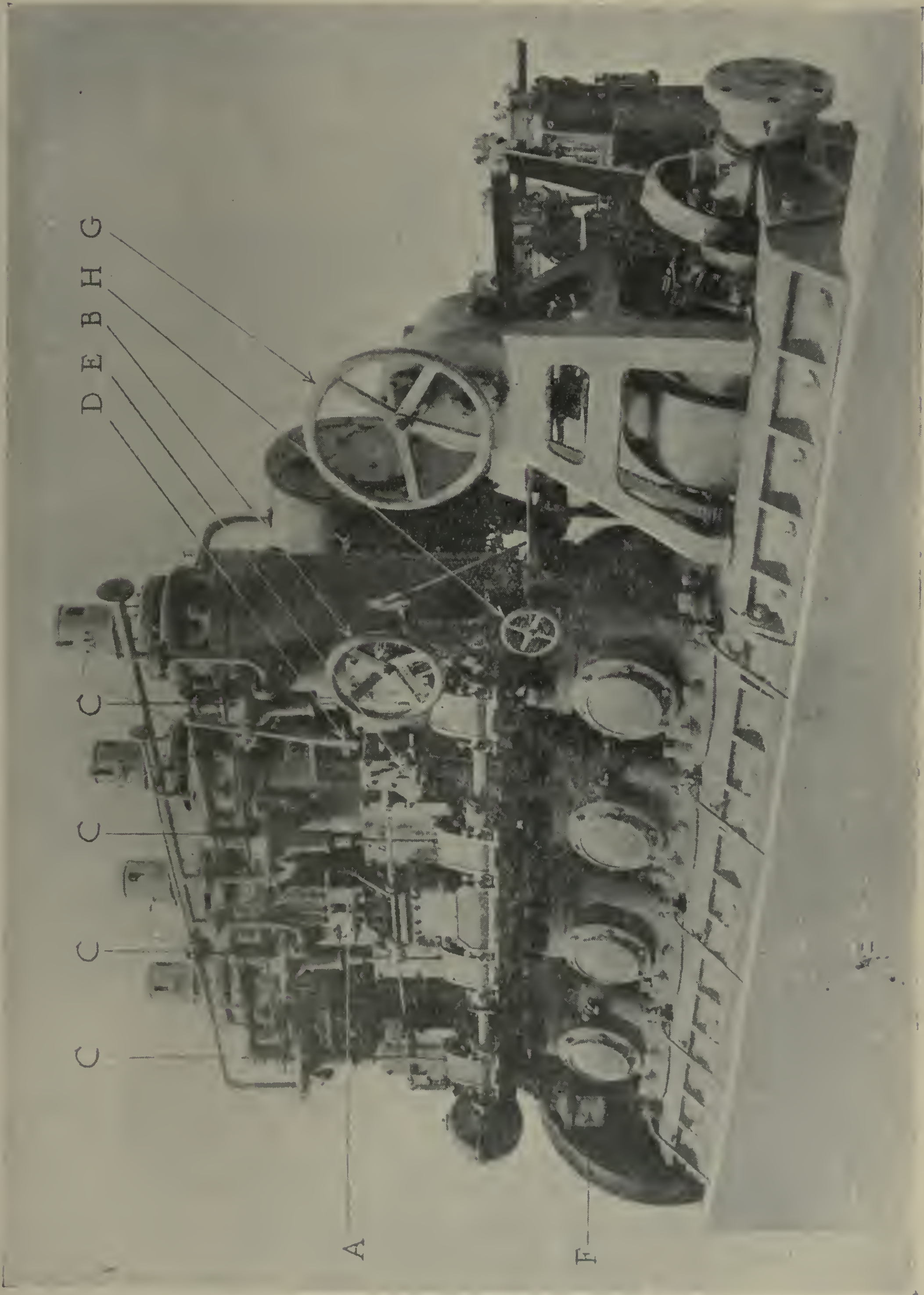


Fig. 8.

direction; this automatically cuts off the fuel pumps and puts the astern air valve cams under their respective valves and so admits air which slows up the engine and starts it turning in the reverse direction. The wheel (B) is then returned to its normal position, which cuts off the air and again puts the fuel pumps into action and the engine continues running astern on fuel.

It is of course necessary in an engine of this class to have a reliable supply of compressed air, and for this it is advisable that such installations should be equipped with an auxiliary air compressor, although of course, should the air supply run somewhat low, the engine may be declutched by turning the wheel (G) when the reduction of the load enables it to be reversed with a much lower pressure than when connected to the propeller.

Large mercantile marine Diesel engines require a good deal more elaborate mechanism for reversing than the surface ignition engine. In the four-cycle engine it is even more complex than in the two-cycle type owing to the former having four valves—exhaust, inlet, fuel and starting—of which the setting has to be altered when passing from ahead to astern. The two-cycle type has only two valves which require re-setting when reversing—the starting and fuel valves—as in this type the exhaust gases escape from ports uncovered by the piston at the lower part of the stroke, and in most designs have the scavenge air admitted by ports on the opposite side of the cylinder in a similar manner to the surface ignition engine.

Exceptions in the two-cycle arrangement are shown in the Polar Diesel type of engines, which have only the fuel valve in the cylinder head and in which the starting and reversal of motion is carried out in the two-cylinder scavenge air pump or air compressor which is coupled to the forward end of the crankshaft. The two cranks of this pump are set at 90 degrees and for starting and reversing it operates like a steam engine through using compressed air.

With engines of 1,000 H.P. or more, ordinary hand operated reversing mechanism would be too slow and heavy, as in steam engine practice. So on large engines, servo-motors worked by compressed air are generally used for moving the camshaft and lifting the valve levers, the servo-motor acting in the same capacity as the reversing engine in steam reciprocating machinery.

Fig. 9 show a type of reversing gear used by Messrs. Vickers Limited, on their *Narragansett* type mercantile Diesel engines with six cylinders. This engine is of the four-stroke type with solid injection of fuel, and is somewhat similar to those fitted on the later submarines which were equipped with Vickers' solid injection Diesel engines. In passing, it may be men-

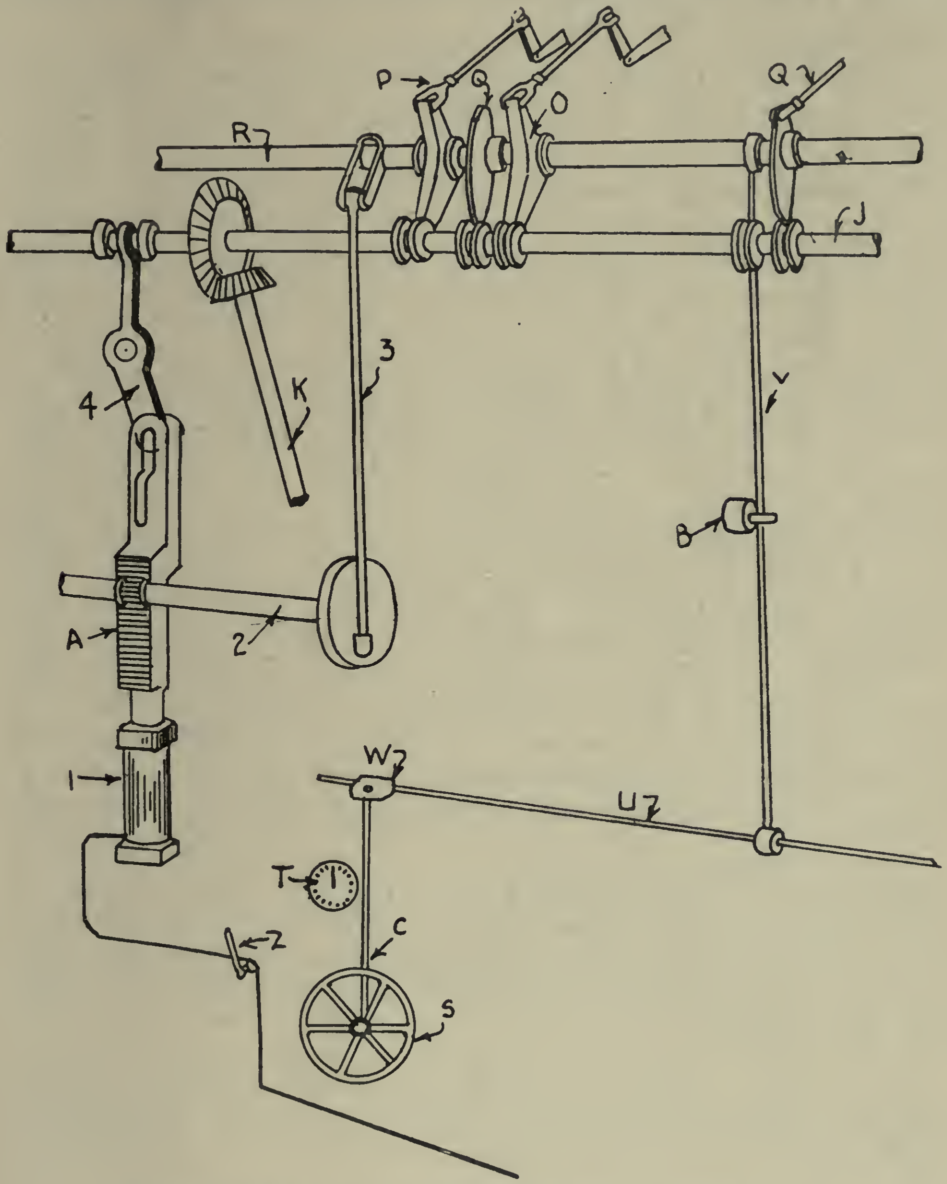


DIAGRAM OF VICKER'S DIESEL REVERSING SYSTEM.
AS FITTED ON M. S. "NARRAGANSETT"

Fig. 9.

tioned that the majority of the Vickers' solid injection engines, which were fitted in the submarines, are of the non-reversible type, the reversing of the vessel being carried out by the electric motors, which are, of course, disconnected from the main engines for the purpose.

If the telegraph rings for full astern when the engines are running slow ahead the following is the sequence of operations:—

The manœuvring wheel (S) is turned by hand, which through gearing (W) rotates the shaft (U) and so through the rod (V) lifts the fuel valve lever (Q) which is mounted on an eccentric fulcrum sleeve, off its cam so that the fuel valve no longer opens. This same operation is carried out with each pair of cylinders in turn until all the fuel valves are out of action when the indicator (T) which is geared off the spindle (C) points to "stop," and the engine will then only revolve due to the trailing of the propeller. The reversing lever (Z) is then pulled over to the front, which admits compressed air to the servo-motor (1), the piston of which is connected to the toothed rack (A). The first movement of the servo-motor pushes the rack upwards and through the pinion shewn, partly rotates the shaft (2), which in turn rotates the manœuvring shaft (R) through the rod (3) and so lifts the exhaust valve lever (O) and the inlet valve lever (P) off their respective cams, by means of the eccentric fulcra, upon which they are mounted on the shaft (R); meanwhile the air valve tappets are clear of their cams till air pressure is applied and no disengaging gear is required.

Further upward movement of the rack (A) moves the cam-shaft (J) to the right, through the crank (4), and so brings the astern cams under the levers (P), (O) and (Q). Continued upward motion of the rack (A) rotates the manœuvring shaft (R) back to its running position and returns the levers on to their respective cams.

All valves are now set in position for astern running, and the next operation is to turn the wheel (S) by hand through the following consecutive positions, as shown by the indicator (T):—

First position: All cylinders receiving compressed air.

Second position: Two cylinders (one pair) receiving fuel, the rotation of the shaft (U) through the vertical rod (V) having brought the fuel valve levers down on to their cams, the others still receiving compressed air.

Third position: Two pairs of cylinders running on fuel and the other pair receiving air.

Fourth and running position: All cylinders on fuel and the air shut off.

The fuel admitted is then increased until the desired speed is attained running astern.

For again putting the engine into the ahead direction, the wheel (S) is turned further to a "stop" position, when the fuel valve levers are out of action. Pressure is admitted to the "ahead" side of the servo-motor, so that the rack moves downwards and so moves the camshaft to the left. The wheel (S) is then put through the same sequence of operations as for starting "astern."

The camshaft (J) is driven through a vertical shaft (K) and skew gearing from the crankshaft, the large bevel wheel being mounted on a sleeve, through which the camshaft slides.

The sketch only shows the mechanism in a diagrammatic form for one cylinder only, and all brackets and bearings have been left out for the sake of simplicity; the fuel valve lever for the next cylinder to the right, has however, been shown separately, so that its operation can be seen more clearly than as shown between the two valve levers in the left hand cylinder.

It is of the greatest importance for successful manœuvring of a large engine like this, that the supply of air should be reliable, as not only has the engine to be started on air, but air has to be used for slowing down the engine by its cushioning action as in a steam engine.

As previously mentioned, the two-stroke cycle Diesel engine is somewhat more simple to reverse than the four-stroke type, owing to there being fewer valves in the cylinder head. The scavenge air is admitted through ports uncovered at the lower end of the piston stroke, and the exhaust also passes out through ports on the opposite side of the cylinder, in a similar manner to that usual in the surface ignition type of engine. It will therefore be seen that owing to the absence of exhaust valves and air inlet valves, the reversing mechanism is considerably simplified. An example of this is shown in the Polar Diesel engine, the reversing gear of which is shown in Fig. 10. This system is used by the Atlas Diesel Co., on their two-stroke engines up to 500 B.H.P.

In this type of engine, a scavenge air pump (A) is required, which supplies air to assist in expelling the exhaust gases from

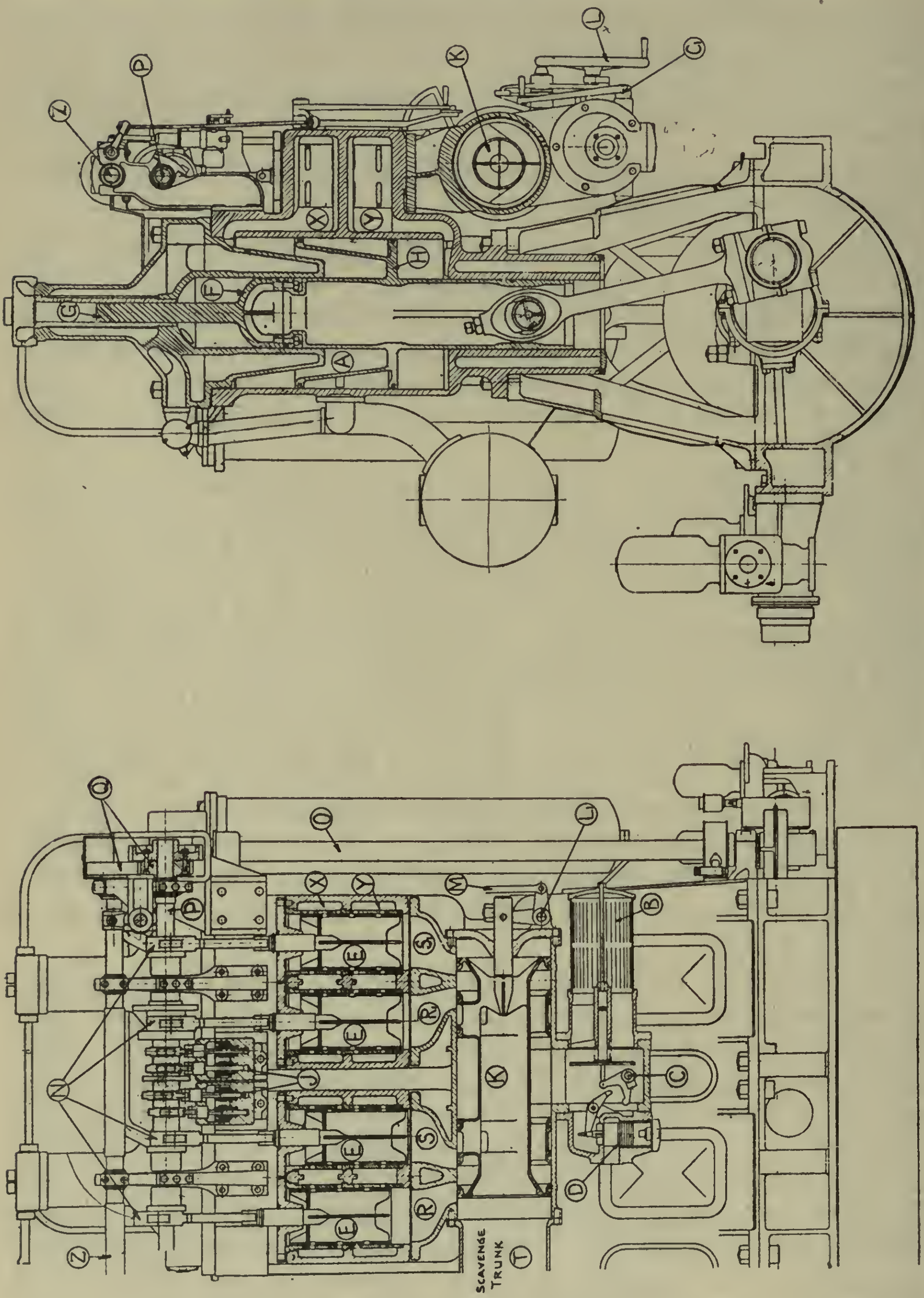


Fig. 10.

the cylinder as is done in the surface ignition engine, by the air compression in the crankcase. For reversing purposes, this air pump becomes a manœuvring engine. There are two pistons and cylinders, the cranks of which are set at 90 degrees to each other, so that this pump acts exactly like a two-cylinder double acting steam engine, and there is no dead centre. The crankshaft of the pump is bolted to the forward end of the main crankshaft. In this particular case the engine has four working cylinders and two pump cylinders. For starting up, the slots (B) which, when the engine is running, are used as the air intake of the pump, are shut off, and in the same operation of the lever mechanism (C), air is admitted from the starting air receiver through the valve (D) to an arrangement of piston or sleeve valves (E) which convert the scavenge pump into a manœuvring engine in a similar way to a double acting steam engine, by admitting air to both sides of the piston.

Before compressed air can be admitted, the fuel pumps are cut out of action, and fuel cannot be delivered to the engine again until the direction of rotation has been changed.

Referring to the diagrams, the action of changing from ahead to astern is as follows:—

The manœuvring wheel (L) is turned to the position of "stop," and through the vertical rod (M) cuts the fuel pumps (J) out of action, and at the same time moves the piston-reversing valve (K) over to its astern position to admit air to the sleeve valves (E) and so the air passes to the proper side of the piston for the direction in which it is required to start the engine.

It will probably be better understood if the valve (K) is likened to the reversing valve of some steam capstans, in which steam is admitted through the exhaust ports and exhausted through the steam ports and *vice versa*, according to the direction in which the engine has to run.

The valves (E) receive their motion from eccentrics (N), mounted on the camshaft (P) which also operates the fuel pumps and on which the governor is mounted at the left hand end. This camshaft is driven off the crankshaft by the vertical shaft (O) through gearing (Q).

It must be understood that when the piston valve (K) opens the ports (R) (R_1) to admit air, the ports (S) (S_1) are opened to the exhaust, and this will start the engine revolving in the direction required. On the other hand when the piston valve is moved in the opposite direction, the ports (S) (S_1) are opened to the admission of air and the ports (R) (R_1) are opened to the exhaust, the air being exhausted into the scavenge air trunk (T) and passed out over the engine pistons in the usual manner, which of course assists in scavenging the cylinders during the time of reversing.

It is also to be noted that the manœuvring engine still acts in its capacity of air compressor until the kinetic energy of the flywheel is exhausted so that as this is still charging the reservoirs there is small loss of air in starting the engine again; at the same time it acts like steam used in a steam engine for cushioning, and so helps to stop the propeller revolving.

The manœuvring wheel is then turned to the astern position, which, by means of the vertical rod (M) moves the camshaft (Z) endways, so altering the timing of the fuel injection valves in the centre of the head of the working cylinders (not shown in diagram).

The lever (C_1) is then moved to the right which turns the spindle (C) and so closes the air intake (B) of the scavenge pump and at the end of its movement opens the valve (D) and so admits air from the starting reservoirs.

Each manœuvring cylinder has two distributing valves (E) (E), which, as explained above, are in turn used for admission and exhaust. When the valve (K) has been moved over to its proper position the air is distributed to the admission valves (E), which admit air into the cylinder in exactly the same way as the slide valve of a steam engine. The crankshaft then starts revolving and the working cylinders commence firing in the reverse direction, and the lever (C_1) is then released and returns to its normal position, shutting off the compressed air and opening the scavenge pump air intake. The engine is then working under normal conditions.

By means of a special arrangement on the shaft driving the fuel pumps and connected with the governor, immediately the fuel is by-passed, when the manœuvring wheel is turned to "Stop," each pump then has one charge of oil left in it and this is injected into the cylinder before stopping, which ensures that no oil can accidently enter the working cylinder at the moment the engine is changing its direction of rotation, and as soon as the engine is actually reversed on compressed air, the fuel pumps automatically commence to inject oil, so that the changing from ahead to astern is carried out rapidly and is only dependent on the handling of the lever (C_1) and the manœuvring wheel (L).

The scavenge pump is also utilised as a two-stage air compressor, as shown in the diagram, by means of an extension and the scavenge piston trunk (H) in the form of two stepped pistons (F) and (G), and these supply the blast air for injection, and also air for manœuvring purposes. This of course, is distinct from the scavenge air which is supplied at a low pressure by the main portion of the piston trunk (H) which is larger in diameter than the compressor piston.

As mentioned in the introductory remarks, the foregoing particulars are merely descriptions; for instructional purposes, however, fuller particulars and notes on working, points to be observed whilst running and hints on overhauling will no doubt be of more general interest, and it is hoped to deal with them fully in some of the subsequent papers of this series.

A MEMBER: The author says there is difficulty, and no doubt there is difficulty in the oil engine with reversing, and he gives four systems for reversing oil engines. The oil engine's salvation is continuous running, so if you have a continuous running engine you need to adopt means to reverse between the prime mover and the propeller or the blades, and I cannot say I am in favour of the propeller blade reversing. I take it that the air pressure system, with compressed air, would be the quickest, easiest, and safest method of reversing the two-cycle engine.

A MEMBER: It is common knowledge to all Marine Engineers that it very often occurs that when your engines are going full speed ahead you have suddenly to reverse to full speed astern. In the author's opinion is it possible that the internal combustion engine will reverse equally as quick as the steam engine? Will a pilot take an internal combustion-engined vessel up a river for the same cost as he will a vessel driven by a steam engine when he knows he can in the one instance get a reversal of the machinery in less than 30 seconds, and in the other case in probably not less than 3 or 4 minutes?

A MEMBER: Speaking generally on the making of these machines and the running of them, I do not see any difficulty in reversing the oil engine. The author says it has to be very carefully done, and in some cases it is nearly impossible, but I do not see any great difficulty about it. There are many engines built where a half-time shaft is used; on this shaft are two sets of cams—one for ahead and the other for astern. To reverse this engine the half-time shaft is lowered a little, then pushed on end, then raised again, causing the other set of cams to act on the valves, which would give a difference in rotation. If your engine has been running full speed ahead and the ship doing her usual speed I don't think you would be able to reverse the oil engine right away without some brake action to stop the engine revolving ahead. Then, when this had been done some time, reverse engine, then apply compressed air to get a move until it would be safe to admit explosive mixture.

A MEMBER: Part of my experience has been in the running of submarine engines during the war. Although we had reversing gear fitted in some boats it was rarely used; the electric motors were found to be more convenient. I had the pleasure of helping to bring one of the German surrendered U boats into harbour, under her own power. I managed to get the engines going and she ran well. We did not know anything about the engine, but when the reversing lever was pulled over, the engine quietly reversed without any trouble. This says a lot for the design of the reversing gear and the whole engine. It requires a considerable amount of running experience on most types before the engine can be manœuvred without a hitch.

In regard to pre-ignition reversing, could the author tell us what is the average piston speed on the final upward stroke before reversal, and what pressure is usual? If it is possible to have a variation of speeds at which reversal will take place, I think it is liable to strain the crankshaft. The only Bolinder type engine I had to do with was in a large picket boat. The engine went astern every time and we had no difficulty at all.

A MEMBER: In considering the suitability of the type of reverse gear for a particular oil engine there is one feature which should be kept in mind, and that is, that there is no single system that can be used generally for all sizes. A reverse system which would be suitable to large engines of say one thousand horse-power or over would probably not be a commercial proposition if applied to a small engine of say fifty to sixty horse-power. With the steam engine you have a system of reverse which applies practically to any size. The Stephenson link motion is just as applicable commercially to small as to large engines, and the relative cost in each case is not so wonderfully different. But with the oil engine a system which would be applicable in the case of a large engine would probably be quite out of the question commercially in the case of a small engine. Apart from the question of first cost, there would be the extra complication of the machinery which you would be placing in the hands of semi-skilled men. You do not have fully qualified engineers in charge of the smaller engines, and that makes another reason for adopting a simpler arrangement for reversing than in the case of the larger engines. These considerations, I think, do away with the question as to which is the most reliable method, because what is reliable in small engines would not be reliable in large engines. The large Diesel engine as now turned out by well-known firms for ocean-going vessels is started, reversed and manœuvred by compressed air, and the manœuvring of these engines is absolutely reliable. These large engines are coupled direct to the propeller without the intervention of any clutch, reverse gearing or other loose parts, a very favourable feature from the engineer's point of view, apart altogether from the practicability of designing and operating a clutch to carry the large powers coming into common use. As long as a sufficient supply of compressed air is maintained in the reservoirs for manœuvring there is no doubt of the engine responding to what is required of it, either with the two or the four-stroke cycle engine.

A Member: From remarks that have been made regarding the usual way reversing is required to be done on large engines at sea, the only thing you appear to require is to have a separate compressor driven by some means so that there is no difficulty about keeping an ample amount of compressed air to reverse when necessary.

The Chairman: I quite sympathise in the remarks made about the difficulty of reversing. It must appeal to every man accustomed to use steam engines. The steam engine works on a cycle and repeats itself practically at every stroke of the engine. Most internal combustion engines require two or four strokes for a complete cycle. Take the case of the four-stroke cycle. The sequence of events is (1) Suction (2) Compression (3) Expansion (4) Exhaust; if this sequence of events follow in the reverse order, we should have (1) Exhaust (2) Expansion (3) Compression (4) Suction; in addition, since the charge is fired at about the dead centre at the commencement of (3) in the ordinary method of working, the firing when reversed would be at the commencement of compression. Further compression would follow suction, and generally everything when reversed as regards the valve action would be out of phase and the final result, trouble. It is therefore impossible to reverse such an engine by simply turning backwards with the same set of cams. The cam has been used for operating the valve gear of the steam engine, also as a reversing gear it has been tried over and over again. There have been some notable attempts, but in every case there has been a return to the Stephenson link gear. Another difficulty occurs when the engine drives the propeller through a clutch. I suppose the most familiar example that I can quote is the motor-car. If you are going to reverse or change speed you remember you must de-clutch before you can do anything at all. When you put in the clutch aftewards, it is easy for the novice to stop his engine altogether. Comparing a high-powered with a low-power engine, the torques exerted in the two cases are associated with different angular velocities, and the effective arm of the couple will be very different.

The formula for calculating horse-power in general use is:—

$$\frac{P\,L\,A\,N}{33{,}000}.$$

This may be written

$$(P \times A) \times (L \times N)$$

but P × A = Total load on piston, whilst $L = 2R$, *i.e.*, 2 × crank and N = angular velocity of crank measured in strokes per minute.

It is evident then that this formula may be rewritten in the form

$$\frac{T \cdot \Theta}{33{,}000};$$

where T is the mean value of the twisting moment, and Θ is the total angle turned through in a minute. Of course $T = F\,R$ where F = force, and R = radius of arm at which F is effective. So that

$$\text{H.P.} = \frac{F.R.\Theta}{33{,}000};$$

Take the case of an engine of 40 H.P. at 700 rev. per min. then $\Theta = 2\pi \times 700 = 4396$, so that

$$F.R. = \frac{33{,}000 \times 40}{4396} = \underline{300}$$

Again take an engine developing 4,000 H.P. at 70 revs. per min. then

$$F.R. = \frac{33{,}000 \times 4.000}{4396} = \underline{\underline{30{,}000.}}$$

Take for the first engine a clutch of the Weston or multiple plate type, then if R = 6 inches about 20 plates would be required, a quite possible arrangement, but in the case of the second example, if R was kept at 6 inches some 4,000 plates would be necessary! the end pressure in each case being about 100 pounds. If the number of plates be reduced to 100, the radius of the clutch would have to be 10 feet, and a clutch exceeding 20 feet dia. would certainly be an addition to the engine-room plant that could scarcely escape comment. Clearly a clutch is inadmissible except for very moderate powered engines.

Another trouble is that due to the interia of the parts. In a marine engine of course this is small, being outside the engine itself, only the mass of shafting and propeller, although one must not forget the effect of the water upon the propeller.

Altogether it would appear that except for low-powered units, all forms of gearing, clutches, loose bladed propellers are quite unsuitable, and that compressed air is probably the best scheme of reversing gear, with a double set of cams for operating the valves in any engine developing considerable power. Electrical starting gears are of course in use and present some advantages, but experience at present appears to point to compressed air as the most suitable method.

A Member: It seems to me from the lecture that one of the main difficulties with an oil engine that is not fitted with a clutch to go astern is to bring it to a standstill from its ahead motion. May I ask: has it ever been suggested that an affair, something like a casing fitted on the outside of a certain part of the shafting with blades fitted to the shaft could be used, and when you require the engine to be brought to a standstill have a large valve operating from the side of the ship, which would open, and the sea-water would be thrown against those blades and so assist to bring the engine to a standstill? It is only a crude suggestion on my part, and I do not know whether it has ever been thought of or suggested.

A Member: I should like to add that the flexibility of the larger size of Diesel engine is equal in all respects to that of the steam engine with one exception, in reversing from full ahead to full astern. I tried on several occasions to reverse a large Diesel from full ahead but the engine was always sluggish in responding, but this is a feature which I have no doubt will be overcome in the near future.

The Author: The question was asked whether pilots would come up the river with an engine that took longer than 30 seconds to reverse, but the member did not say what size engine he had in his mind, as obviously the size has much to do with the time required. I can tell him of many oil engines that will reverse more quickly than steam engines, give demonstrations with engines up to 300 h.p. that will reverse in $12\frac{1}{2}$ seconds, and can tell him of 100 h.p. Bolinder engines that will reverse from full ahead to full astern in the incredible short time of $8\frac{1}{2}$ seconds. Oil engines will reverse as quick, if not quicker than steam engines of the same power. I was asked to give my opinion as to the most reliable method of reversing, a somewhat difficult subject as I may be considered prejudiced; although originally a steam man, I am also an oil engine man. For very small vessels it does not matter what method you use to reverse, when you come to large engines you must rely, as several of the gentlemen have mentioned this evening, upon air reversing. It is not advisable in my view to have a clutch for anything over 500 horse-power. In intermediate sizes, a difficulty arises, makers who do not have direct reversible gear claim that reverse gears are better. Personally, after an experience of several hundreds of engines, I prefer the direct reverse, because it is much simpler and generally as absolutely reliable as a steam engine. But you must know first of all its method of adjustment and its idiosyncrasies. About eleven years ago I was very sceptical, until I saw the engine reverse. I speak of the Bolinder engine. A test was carried out and three men were put on to take it in turns in reversing, and at the end of the series of tests, lasting several hours, the men were very tired, but the engine never failed once to reverse, and I was convinced.

In the case of reverse gear, if you get a good reverse gear the method is reliable, but I submit it wants more careful attention. You may put over your lever too quickly, and it may cause trouble, just as in a motor-car; you may say "what novices!" but you do not generally put such men in charge of a marine oil engine. You are more likely to do damage with reverse gear, but in any form of direct reverse it is almost impossible to do damage to the engine. There have been cases of breaking reverse gears, but in nearly every case it has been found that the man in charge had lost his head, had got excited, and had pushed the lever before the kinetic energy was off the machine.

A member mentioned the question of flexibility of steam engines, and continuous running. I claim that for engines of 200 to 300 horse-power the oil engine is equally flexible. If the point is one of starting and running very slowly and quickly speeding up, the oil engine is at a disadvantage, but if it is for general handling purposes for reversing, my experience, not only with the engine I have mentioned, but on several other engines, is that they are much more flexible than steam. With steam engines with 180 lbs. boiler pressure I have reversed from full ahead to full astern without shutting off steam; that is a very severe test. In the case of the oil engine you can do that with one lever, you must however, first de-clutch or slow down, unless you are using air, then I doubt if the small valves of the Diesel or four-stroke cycle oil engine will stand such a shock. With regard to the revolutions of the engine, a member raised a question. Well, the feature of the Bolinder engine is that you ease down first automatically. If you have it running at 300 r.p.m. it will ease down to almost a stop, so that the shock or torque is even less than in a steam engine. The torque cannot be considerable if you are reduced to a few revolutions. In the case of an engine with a clutch you have only to overcome the flywheel, and I doubt if that is as great as overcoming the propeller, which is running in the opposite direction.

A member referred to the liability of pre-ignition reversing to strain the crankshaft. It is possible to have small variations of piston speed at which the reversing charge of oil is injected, but in the case of the Bolinder it cannot take place at anything but a very slow piston speed. Referring to Fig. 6 the reversing tappet T at all but the slowest speeds has sufficient inertia to overcome the tension of spring P. This spring is very carefully made and tempered, and it is not possibe to put more than the designed tension on it by tightening the nuts V, as a shoulder prevents this. The tension which is only adjustable within small limits will only overcome the inertia of tappet T just before the engine stops. The piston is brought to rest quite gently as the kinetic energy of flywheel and moving parts has dropped to so low a value, that the ignited pre-ignition charge is only partly compressed, so that the stress on shaft is not more than normal. This is proved by the large numbers of Bolinder engines installed in tugs, river and canal craft where reversing is very frequent, which have been running for years without breakages of crankshafts. Also other makers have followed the Bolinder after many years' experience.

Regarding the reversible Diesel engines on the German submarines. Whilst at the Admiralty I had the opportunity of seeing some of these, and as there was no doubt they were almost perfect in reversing, and for this reason we have to bear them in mind when considering this important question. As far as this class of machinery is concerned, there is no doubt that during the war the Germans took the lead, as is proved by the excellent results which were obtained by the submarine engines under the arduous conditions in which the Germans conducted the submarine campaign.

Some mention was made in regard to the commercial aspect. It is very difficult for me to deal in this paper with all I could say on this matter. Undoubtedly the cost of working the oil engine is much less than in the case of steam. I have in my mind a case on the Thames where a 60 horse-power crude oil engine was fitted that gave some extraordinary results. It was working from London Bridge to the Albert Docks in 1915, and the total week's consumption of fuel oil over a period of twelve weeks worked out at 9/6 per week; these figures were supplied by the owners, and were considered by them so satisfactory that they gave an order for an oil engine tug, the *Grove Place*, of 160 b.h.p. The firm own 9 or 10 tugs, and the saving in the *Grove Place* was so astonishing that at the fall of 1919 they came to the conclusion that they were saving in running costs about £1,000 per annum on the oil engine boat, as against their steam tugs of similar power. The result was that they placed an order for two larger tugs of 350 horse-power with the intention after delivery to gradually dispose of their steam craft and go in for further oil-engined tugs. It is the more astonishing when you think of the trying conditions on the Thames, and the imperative necessity of economy and reliability in handling and reversing in the most crowded waterway in the world. That is the considered judgment of an important firm who have worked both oil and steam-engined vessels for some years. I put that down, not only to the continuous running economy of the oil engine, but to the heavy stand-by losses of the steam engines. It would not be quite fair to apply these remarks in full to any sea-going or coasting vessel.

A question was asked by a member as to the design of the clutch for reversing; I am not quite sure whether he had in his mind reversing vanes. That form has been used for small powers, but I doubt whether it would be applicable to marine jobs of average power. It reduces shock considerably, but I have never heard of it being applied to anything greater than a small launch. I have heard of several types of hydraulic reversing, but I have never seen it developed to any considerable proportions.

The question of reliability of steam *v.* oil was mentioned, and as already explained, I undoubtedly favour the oil engine; steam boilers will wear out, tubes will give out, and then there are the manhole and sight-hole doors and fittings, bunker sides, ash guard plates, gauge glass and fittings, steam and check valves, etc. In an oil-engined vessel you have none of these troubles.

With a good make of engine I would say an oil engine is every bit as reliable as a steam engine.

SESSION 1921-22.

President: SIR JOSEPH P. MACLAY, BART.

The Open-Fronted Surface Ignition Engine.

BY F. G. BUTT-GOW (Member).

Read Tuesday April 12, 1921.

THE modern Semi-Diesel, Hot Bulb or Surface Ignition Engine has not gone through so many radical changes of construction as the marine steam engine did in its early days, and in its present day construction it very much resembles the engine from which it probably originated—the Two-stroke, or perhaps more correctly, Two-cycle.

This engine depended on electric ignition. One of the first marine oil engines to use the hot bulb system of ignition was the Mietz and Weiss; and with very few exceptions marine oil engines of a moderate compression—that is 200 to 250 lbs. and depending on a hot bulb or plate for ignition—are still built in the same form as when originally made.

In Holland, which is one of the first countries to employ the marine oil engine to any great extent many barges were fitted with horizontal oil engines using tube ignition; but this I think was for the same reason that the horizontal engine, either trunk or return connecting rod, was in use for a considerable time with steam vessels, *i.e.*, on account of head room, and also owing to the fact that the bulk of oil engines as made then were horizontal. With few exceptions, some of which are illustrated, the Semi-Diesel or surface ignition, termed the closed, has always been built as a closed crankcase engine. It is true that one Two-cycle engine was put on the market in the smaller sizes with the compression space separated from the cylinder and fitted with a piston rod and stepped-piston. In the larger sizes with four-cylinders, these were made with an open crankcase and fitted with piston rods and guides, compression taking place as in the smaller engines.

However, as the majority of the firms who took up the marine oil engine had been engaged in land work they built marine engines as they had been accustomed to for land use, which may account for the large number still made with a closed crank case. I was told by a representative of one very old firm of oil engine builders that they did build an open-fronted oil engine, but gave it up owing to the expense of manufacture; this point I will refer to later.

Now that the Semi-Diesel is getting on in years and also in size, the question has arisen "Is the closed crankcase with its defects the best design for a marine oil engine?" It has been said that, if the Semi-Diesel is to be successful in the larger sizes, the closed crankcase must be done away with. I may be unduly favourable to the open engine, but the only points I can see in favour of the closed engine are less height and less weight.

An open engine is naturally higher than a closed engine, but except in smaller sizes the question does not often occur. The extra height, comparing engines of similar horse-power, is from 2ft. to 2ft. 6ins. In a tug boat or auxiliary vessel this extra height can be easily spared.

As to less weight this does not always apply. Various builders of the closed engine and their representatives with whom I have discussed the merits of the two types are willing to admit that the open engine is a good one: their great objection seems that it costs more to build. In the smaller sizes up to, say, 30 or 40 h.p., this may be true; I am not in close enough touch with manufacturing costs to be able to pass an opinion. In the larger sizes I do not think there can be very much difference in the cost of manufacture between the two types. To a firm not engaged in marine steam work, there is no doubt the closed engine appeals; but the majority of Marine Engineers with whom I have discussed the point, favour the open engine, and it has been referred to on several occasions during the various papers that have been read at the Institute. I will now take the two types, a closed engine and an open engine and compare them with one another. In the first type we have:—

Fig. 1. The lower half of the crankcase forming the soleplate, and the upper half of the crankcase forming bottom of cylinder. In a four-cylinder engine with about 80 to 90 h.p. per cylinder, these two castings represent a considerable amount of pattern making, foundry work and machining; the lower part of the crankcase in particular requires a lot of work and also entails, in the larger sizes, special machinery.

THE OPEN-FRONTED SURFACE IGNITION ENGINE.

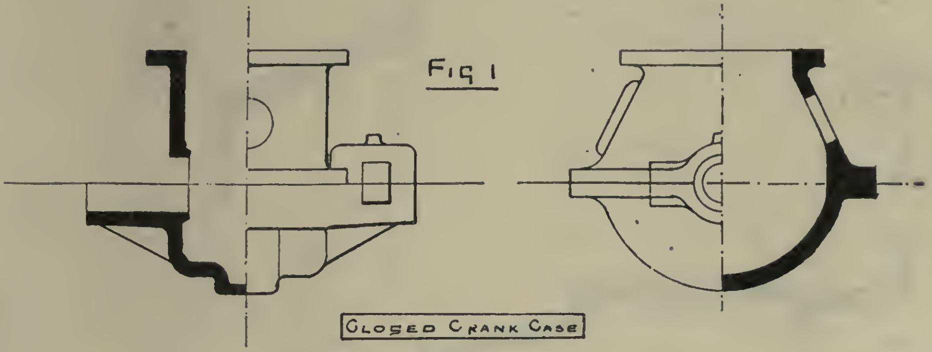
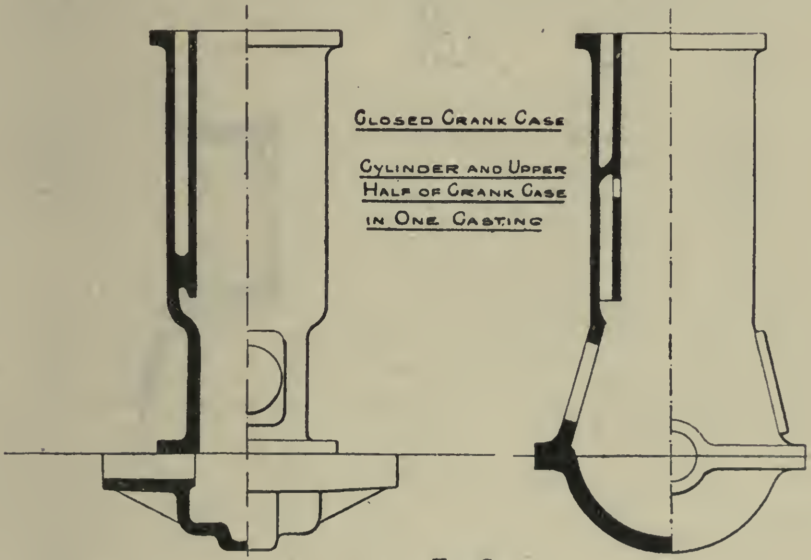


Fig 1

Sole Plate and Upper Half of Crank Case

Closed Crank Case

There are several engines in which the cylinder and upper half of crankcase are made as one casting. This is shown in Fig. 2. As can be seen, this also entails a considerable amount of labour in both the pattern shop and foundry and heavier plant in the machinery. It also means more delay in progress through the shops, as unless special machinery is provided not more than one operation can be carried out at a time, whereas with Fig. 1 the cylinder casting can be boring out whilst the upper half of the crankcase can be on a planer or miller as the case may be.



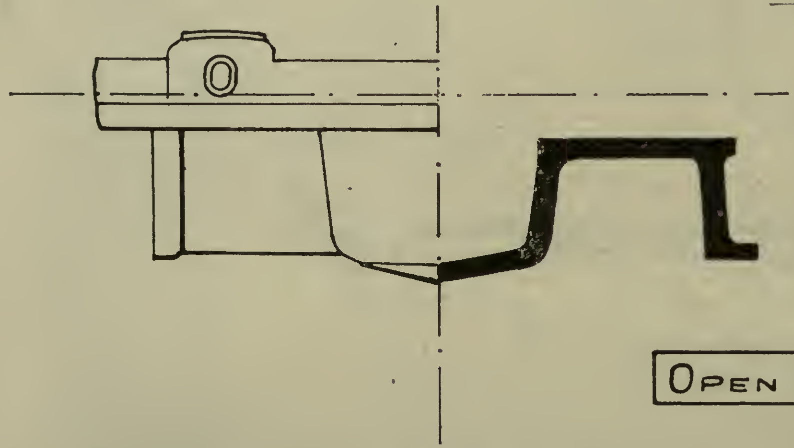
Closed Crank Case

Cylinder and Upper Half of Crank Case in One Casting

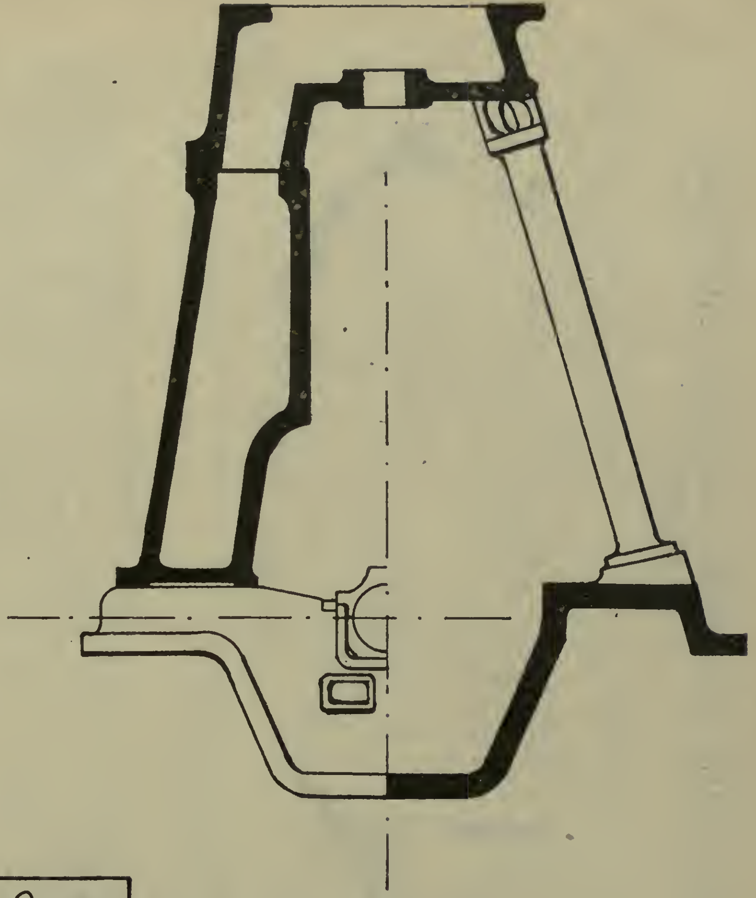
Fig 2.

Fig 3.

OPEN CRANK CASE.



SOLE PLATE.



SOLE PLATE AND BACK COLUMN.

AND FRONT COLUMN

THE OPEN-FRONTED SURFACE IGNITION ENGINE.

The Open Engine.—Coming to the construction of the open engine the most general form is as shown in Fig. 3, consisting of sole plate, back column and front column. The sole plate is very similar to that of a marine steam engine except that in one style of oil engine made of the open type the crank pit is not left open to the bilge, but is arranged with one or more sumps for the used oil to drain to. This does not require so much work as an enclosed engine, no provision being needed for sealing round the crank shaft at the main bearings. These are fitted as in a marine steam engine. In the majority of open engines the front columns are forged and turned, this is not a complicated piece of work, a forging with flanges top and bottom. The back column is a casting as shown. It does not present any difficulties in casting or machining, the pattern is also a simple one. In addition to the back and front columns in one type of engine the upper portion of the compression space is a separate casting, this contains the piston-rod gland and stuffing box and air-inlet valves. There is no reason, however, why this should not be a part of the cylinder as the gland could be bolted on to allow for a boring bar passing through when first the cylinder is bored or should it require boring at some future period.

The cylinders on the two types of engines shown in Figs. 4 and 5, are not very different, possibly that of the closed engine may be slightly longer. The piston on the open engine may be made considerably shorter than that of the enclosed engine as it does not have to perform the duties of a slide in the cylinder walls which serve as guides. The open engine has a piston rod in most cases flanged at the top end for bolting to the piston and with a tapered and screwed end for attachment to the crosshead, most cases flanged at the top end for bolting to the piston and with a tapered and screwed end for attachment to the crosshead, though the crosshead can be made with a single brass, and piston rod secured by taper end and cotter.

The crosshead and slide work in the usual form of open guide, with side strips for astern wear. These, of course, in the case of a non-reversing engine act as retaining strips only, as the guide on the back column face receives the pressure for ahead, and there is no necessity for astern guides. The actual additional parts required for the open engine are piston rod and crosshead with slide and guides.

To go thoroughly into the cost of manufacturing the two types, the open and closed, would take up more time than this paper will permit, but I think it can be seen, from what I have briefly touched on, that the cost of manufacture of the open engine should not be very much more than the closed engine

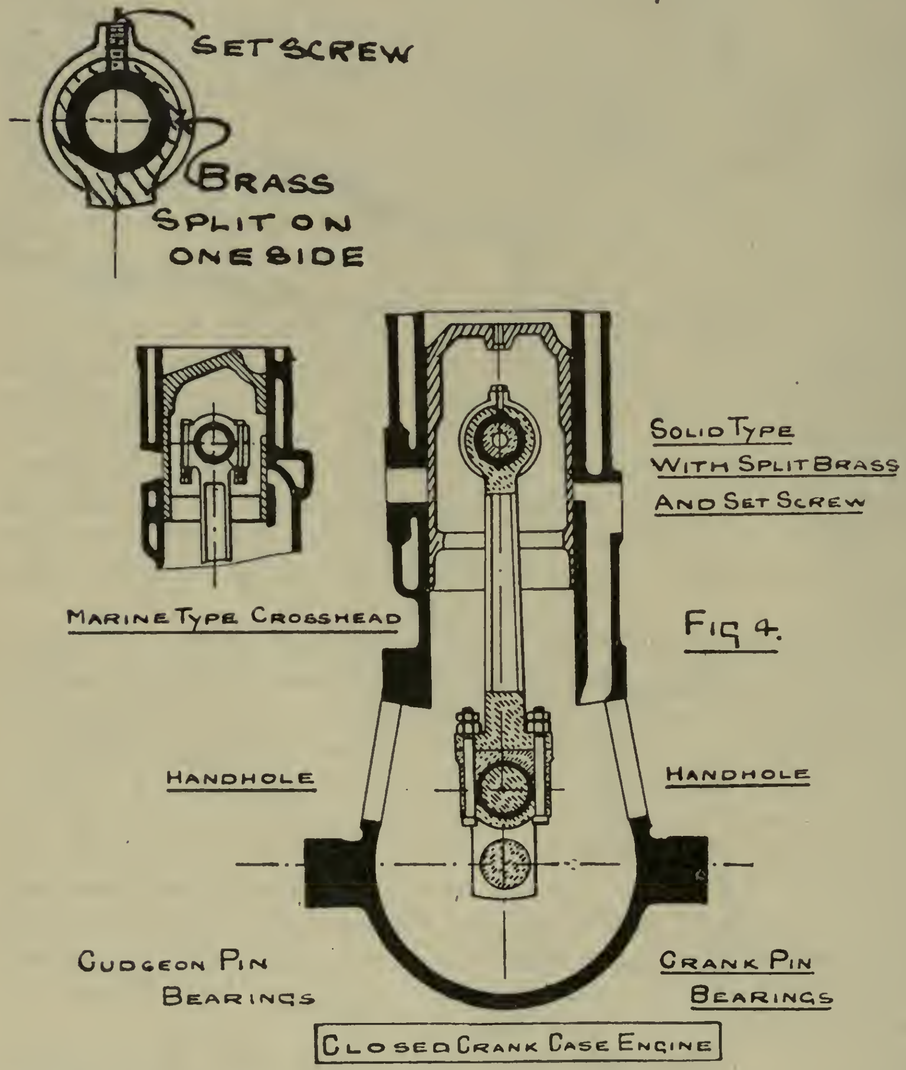
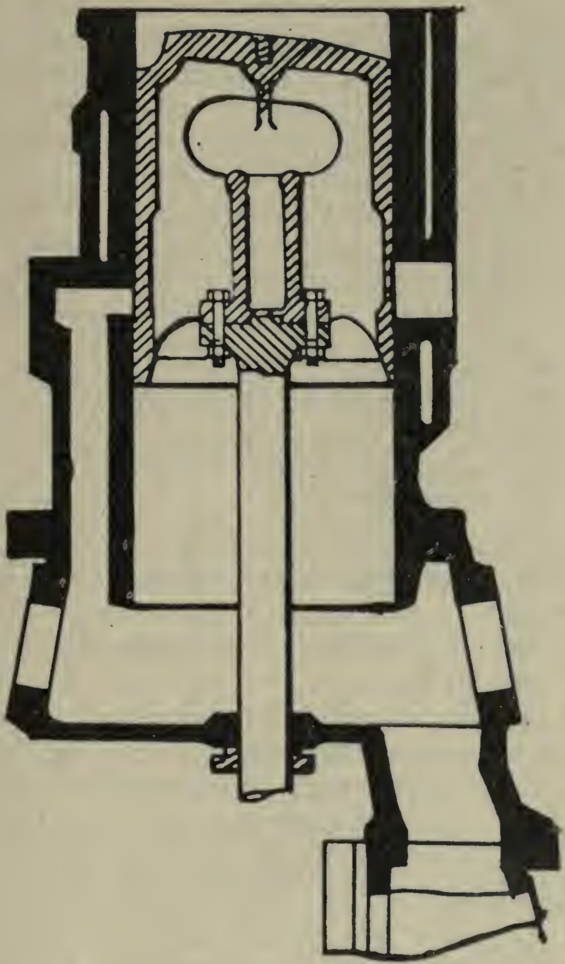


Fig 4.

and in the larger sizes, *i.e.*, 60 to 100 b.h.p. per cylinder, it ought to be less owing to the use of smaller castings and subdivision of parts. More work could be done to the component parts of an engine in the same period and the machine tools need not be so heavy. Another point is the question of lubrication in the closed engine. The fitting of banjo rings and piping requires more care and planning than is necessary on the open engine, the oil piping of which is very similar to a marine steam engine. I will take up the lubrication question more fully in another part of the paper, dealing now with the question of running an open engine as against a closed engine.

Fig 5.



Piston. Open Engine

There have been several articles appearing lately in the *Motor Boat* giving accounts of the running of marine oil engines, and from what can be gathered the bulk of the trouble has been with top and bottom end brasses, loss of compression through main bearings wearing, and reverse gear; this last point I will not go into as the fact of whether an engine is open or closed will not in any way affect the reversing. When reference was made to cylinder wear occurring, it was usually put down to the use of water injection, but as considerable wear is found in closed engines which do not use water, I think the use of a trunk piston, which is necessary with a closed engine, is responsible for a lot of this wear. I think a Marine Engineer used to steam would rather be shipmates with an engine more nearly resembling a marine steam engine. I do not doubt that there are many interested in oil engines who will think that I am old-fashioned in advocating the open type of oil engine for marine use, but I think the practical man would certainly prefer an engine in which the working parts are in view and can be felt. In a previous paper the use of thermometers in submarine engines was mentioned, but I do not think the average engineer wants to spend his time on watch taking thermometer readings.

In the open engine the main bearings, top and bottom ends and guides are in full view all the time. Light splash-plates are fitted, usually hinged just under the cylinder bottoms and the bottom portion made removable for inspection of bottom ends and main bearings.

In a closed engine the main bearings only are in sight; the crank pin must be to a very great extent taken on trust; as the lubrication to this is in most cases forced and not fitted with sight feed, it must be taken for granted that the oil is going in and doing its work.

I read some time ago that the heat of the crank pin could be judged from the blow back at the air inlet valve, but crank pins have a way of heating up without giving warning. In the case of a closed engine the only guide apart from this is the temperature of the crankcase, the engine slowing down, or possibly pulling up.

In the case of the gudgeon pin, I have not heard of any way by which this can be gauged as to how it is running except sound. I do know, in my two years' experience in charge of motor-craft in France, the percentage of repairs to gudgeon pins and brasses was very high.

Lubrication.—On the closed engine the lubrication is generally effected by some system of forced lubrication, though with the exception of the main-bearings, I must say it is rather a stretch of imagination to call it forced, on many engines. The crank-pin brasses are usually supplied with oil by means of banjo-rings or centrifugal lubrication. The gudgeon-pin brasses are lubricated from the supply of oil which is forced through the cylinder walls, this oil passes to the pin and brasses through a hole bored axially through the pin and in connection with a hole at right angles. Some engines are fitted with a scraper to collect the oil.

In an open-fronted engine the lubrication is practically the same as on a marine steam engine; either an ordinary open drip system may be fitted, or a drip with sight feeds open or closed, if desired, forced lubrication by piping as shown by the late Lieut. Windram in his contribution to the Transactions in 1918, page 209. It can be either pressure or gravity as may be preferred. Up to the present I have found the system of lubrication adopted in the Bolnes engine as very reliable and fool-proof. In this engine the oil passes from a receiver fitted with a sight gauge through an open sight feed to the various bearings or sumps, the used oil drains to a sump in the crank-pit, from there it is pumped up through a filter and strainer to the receiver; the amount of oil passing to each bearing can be regulated by the small drip cocks. If it is required to supply any one or more bearings with extra oil it can be poured down the supply pipe of these bearings; on stopping the engine for any considerable time the valve or cock between the various feeds and the receiver may be shut, but if this should not be done no great harm will happen, as the capacity of the crank-pit is greater than that of the receiver the oil will drain to the pit. On starting the engine it will be pumped up again to the receiver and circulate through the various bearings. A very similar plan is now fitted on the Sumner open engine.

As Mr. Warne mentioned in his paper of December 14th, a compromise has to be made over the question of lubrication on the usual (closed) type of Semi-Diesel. If an excess of oil is used it will accumulate in the crankcase, and if care is not taken to keep this drained, considerable damage will be done to the engines if lubricating oil is taken up with the scavenging air.

On the open engine there is no such trouble: more oil can be circulated through the various bearings than on a steam engine, as the excess or used oil will drain to the sump, and from there pass through to the receiver: should the pump break down or choke, it can be seen at once, and if it is not possible to get the pump going, the receiver can be filled by hand and the crank pit drained when necessary with a hand pump.

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It cannot be denied that the closed engine is very heavy on lubricating oil, but this is not so much on the amount of oil that is used as on what is wasted, *i.e.*, drained into the bilges to avoid the possibilities mentioned previously.

A careful driver or engineer would drain off any accumulation of oil at regular intervals, but as in many engines this drain is at the lowest part of the crankcase, it is a simpler matter to leave it open sufficiently to allow the oil to drain out into the bilge. It is not an easy matter to get exact figures of lubricating oil consumption.

Lubricating Oil Consumption.—A correspondent to the *Motor Boat* writes that on a new 90 b.h.p. closed engine the consumption of lubricating oil for a 240 mile journey is 20 gallons. He also mentions having to renew crank-pin brasses every three months.

In a tug fitted with a "Bolnes" open engine of 130 b.h.p. the lubricating oil consumption over a working period of 36 hours was three gallons.

It must also be taken into account that at least 50 per cent. of that three gallons is still fit for service, as it is either in the crank-pit or ready for use in the receiver, as will be understood from the description of the system of lubrication fitted on the "Bolnes."

An engine of rather more power, *i.e.*, 160 b.h.p. of the closed type and working under similar conditions is stated to have had a lubricating oil consumption of 14 gallons in a working period of 48 hours.

On a basis of lubricating oil consumption per b.h.p. per hour the following figures may be of interest.

"Bolnes" 130 b.h.p. ·005 pints per b.h.p. per hour.

Closed Type 160 b.h.p. ·010 pints per b.h.p. per hour.

Closed Type 90 b.h.p. ·52 pints per b.h.p. per hour.

In the case of the 90 b.h.p. engine the working hours have been taken as 34, this is allowing an approximate speed of seven knots per hour, if a higher speed is allowed for, the consumption of lubricating oil will be higher in proportion.

Maintenance.—Running repairs are less on the open type than the closed engine. By reason of the crosshead and guide taking up the side strains, the walls of the cylinders do not tend to wear as in the closed type where the piston rings and cylinder walls are subject to this strain. Also by having a piston rod and connecting rod, the reversal at each revolution is smoother. Any slight wear can be taken up, such as slackness in a crosshead or bottom end, without having to dismantle the engine. The gland through which the piston rod passes requires very little attention, as it only has the compression of the scavenging air, which should not exceed six or seven lbs.

Overhauling.—When it comes to the question of carrying out an overhaul, I do not think anyone can deny that the open engine is very much superior to the closed engine. Starting from the top, cylinder covers and hot bulbs are the same for either type. But when we come to removing a piston, it is quite a different matter. In a closed engine, if it is required to remove the piston for examination, hand hole doors must first be removed, crank pin brasses disconnected, and in most cases piston and connecting rod lifted bodily out. It must also be remembered that all these parts are smothered in oil, and that the space for handling spanners and removing brasses is limited.

Gudgeon-Pin Adjustment.—This is one of the most unsatisfactory features of the closed engine. In some cases no provision is made for adjustment: the connecting rod top end is fitted with a solid brass and a hardened steel gudgeon pin, which is secured to the connecting rod jaws by set screws. Many will say that there are closed engines running that have had little or no trouble, but is the life of a gudgeon bearing anywhere near the life of that of an ordinary marine engine? In many cases on examination it means not only new brasses but a new pin, and it very often happens that the eye of the rod requires boring out. There are many steam engines running now that still have the spare top end brasses bolted up on the bulkhead amongst the spare parts where they have been for ten or fifteen years. But I doubt if there are many closed type oil engines running over five years that have not had at least one set of new gudgeon brasses and possibly pins put in. In cases where the brasses are adjustable, it cannot be said that the adjustment is an easy matter in cases where the brasses are split as shown in Fig. 4 centre view, in this case the adjustment is usually made by a set screw on the top. To adjust this it is necessary to remove the piston and connecting rod, make the required adjustment and replace. It must not be forgotten that this adjustment is not made at the working temperature so that, to be on the safe side, it must be left, if anything, on the slack side. This type requires good workmanship, and a man who can handle a file, to ensure that the two faces bed equally and are brass and brass.

The other method, Fig. 4 top corner, right-hand side, brass split on one side, which is not much in use, is only fit for small brasses, but it comes in very handy in the event of a rod with a solid bush; where no spare bush is available a hole is drilled and tapped and a set screw fitted. It will also be found that after a time, however good the pin fits the eyes of the rod, it will show signs of slackness in some cases so much as to require a new pin and the eyes removed or bored to suit. When in charge of motor repairs during 1917-18-19 I had several cases in which this had to be done. Some engines are fitted with a regular marine type crosshead brass, Fig. 4 shown left-hand side, but this again cannot be properly adjusted without removal as if liners are fitted it is not a very easy matter to remove or replace these, and for reasons of safety it is better to do without them and fit brass to brass.

Crank-Pin Adjustment.—This is an easier matter than the gudgeon pin or top end, but even with hand holes made as large as possible it is not a simple matter, and when it comes to 80 B.H.P. and over on one cylinder, it means a good deal of man-handling and knack, combined with strength: there is not room for working small chain blocks, at any rate not in any that I have seen, and again there is not too much light. Also if it is required to examine a crank pin, or perhaps go over with a stone, it is not everybody's job, or one to be envied.

To sum it all up, in most closed engines to remove a piston you must disconnect crank pin brasses though they may be perfectly satisfactory. To examine the gudgeon pin brasses you must remove piston: this means dismantling cylinder cover, and disconnecting crank pin brasses. To overhaul crank pin bearing in the majority of closed engines you must remove cover in order to hang up the piston and connecting rod. I have not yet seen any provision for doing this in any other way. From this it will be seen that to do any one of three different jobs, the whole or part of one of the remaining two must be either wholly or partly done. I will now take the same three jobs on an open fronted engine.

Removing Piston.—After removing cylinder cover, if the engine has the box type of crosshead, the cotter is knocked out and piston and rod drawn: in the case of the double cross-head, the nut is removed: it will be noticed that neither crosshead nor crank-pin are disturbed. To adjust crosshead and crank-pin brasses, the crosshead can be hung or by a bar across the guide face as is done in ordinary marine steam practice.

Main-Bearings.—These, as now placed in most closed engines, are outside the crankcase and are accessible, but it must be remembered that with age the shaft will be down and loss of compression result; sealing rings require attention, and if it is required to remove the shaft it means removing cylinders and upper half of crankcase.

In an open engine, to remove a crank-shaft does not mean any more than disconnecting and removing top brasses, but with a good quality white metal it should be a long time before this is necessary. At any rate there are no sealing rings to worry about, as the compression is not affected by the shaft being down. To prevent loss of compression on an open engine, it is only necessary to watch the air inlet valves and the piston rod glands, which is not a difficult matter as the pressure on the gland does not exceed seven to ten lbs. So far I have gone into the question of:—

(1) Cost of manufacture of closed engine against cost of open engine.

(2) Running closed engine and open engine.

(3) Overhauling the two types of engine.

(4) Lubrication.

(5) Extra height of open engine.

Note on Fitting Drip Trays.—In an open engine such as the "Bolnes" it is not necessary to fit a tray under the engine as must be done in the closed engine. There is no possibility of lubricating oil accumulating in the bilges. All the oil finding its way to the crank-pit is accounted for by the pump.

I should like to go into the question of working the various pumps required: fuel, cooling, lubricating and air compressing, and bilge. Taking the fuel pump first. In most of the enclosed engines this is driven from an eccentric, the same eccentric also drives cooling and lubricating pumps. These last are principally ratchet driven from the eccentric. As regards the compressed air supply some engines are arranged to do this by means of the exhaust, others are fitted with a compressor driven by another eccentric in addition to that driving fuel pumps, etc.

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There are also several engines having the pumps partly driven from eccentrics, the remainder by an arrangement of gear wheels. In an open-fronted engine one of the simplest ways of working the various pumps is a similar arrangement to that used on a marine steam engine, *i.e.*, by levers from the crosshead as shown in the figure. At the back are fitted two circulating pumps, one bilge pump and air compressor, from another lever in connection with a bell crank the two fuel pumps are driven. The lubricating pumps, engine and cylinder, are also driven from these levers by a system of rods and bell cranks. In the larger engine, of course, the compressor is better as an independent installation, but in engines to 150 B.H.P. the above arrangement has been very successful. The bilge pump can be arranged to use as a cooling pump in case of necessity and up to four cylinders the levers from the crosshead of one cylinder will do all the work required when an independent compressor is fitted.

Spare Parts and Renewals.—The principal working parts of an open engine are so similar to a marine steam engine that overhauls can be carried by an firm of engineers accustomed to steam engine work, with perhaps the exception of the fuel pumps, and though many builders are marine oil engines are in the habit of laying great stress on the extreme accuracy required for these, I have never been able to see that there was anything in these pumps that could not be carried out by the average repair shop working to the usual degree of limits required at the present time.

The heaviest items on the repair list in the closed engine are: pistons, cylinders (re-boring), gudgeon-pin and crank-pin brasses.

In the open engine with piston-rod, connecting-rod and guide the upkeep of these parts should be very little heavier than a marine steam engine.

The most successful Diesel engines are those fitted with crosshead and guides, and I think that if oil engine builders of the semi-Diesel follow on the same lines, they will find it to their advantage and that of the engineers who will have charge.

Since 1912, when the " Bolnes " works started building the open type of oil-engine they have built these engines for all classes of vessels with the most satisfactory results.

In America the Sumner Company have built several sets, all in the larger sizes and all the reports I have heard have been very favourable.

ADDENDA.

Various Types of Open Crank Case Engines.—About 1905 Messrs. Webster and Bickerton brought out a two-cycle or two-stroke engine in which the compression of the slow in-going air and explosive mixture was not carried out in the crankcase. Fig. 6 illustrates this. In the larger engines the principle was

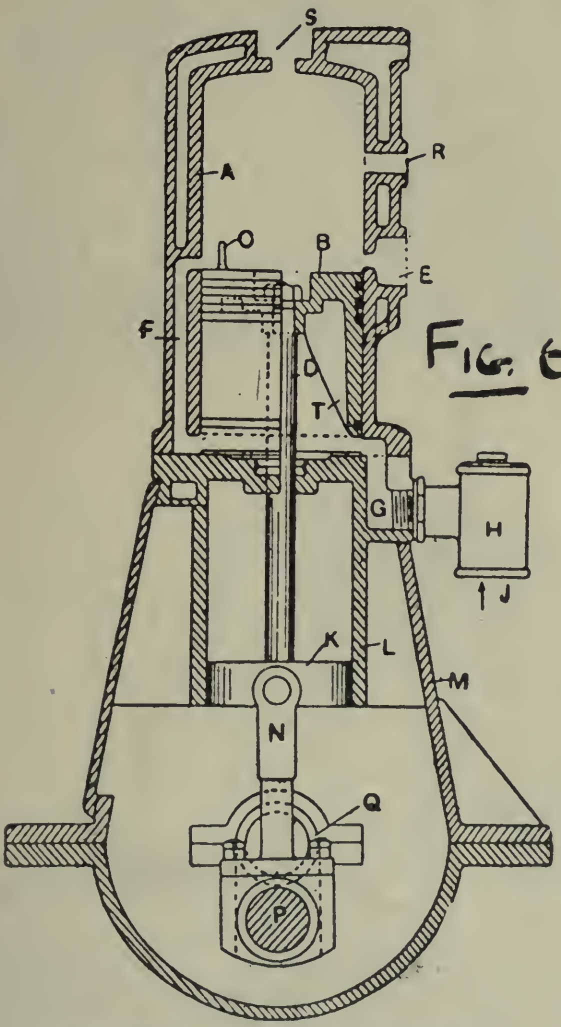


Fig. 6

Section of the "Reliable."

carried still further. Figs. 7 and 8 shows the engine as made in the larger sizes, *i.e.*, 25 and 50 H.P. on four cylinders. I have never seen one of these engines, but I saw one of the first

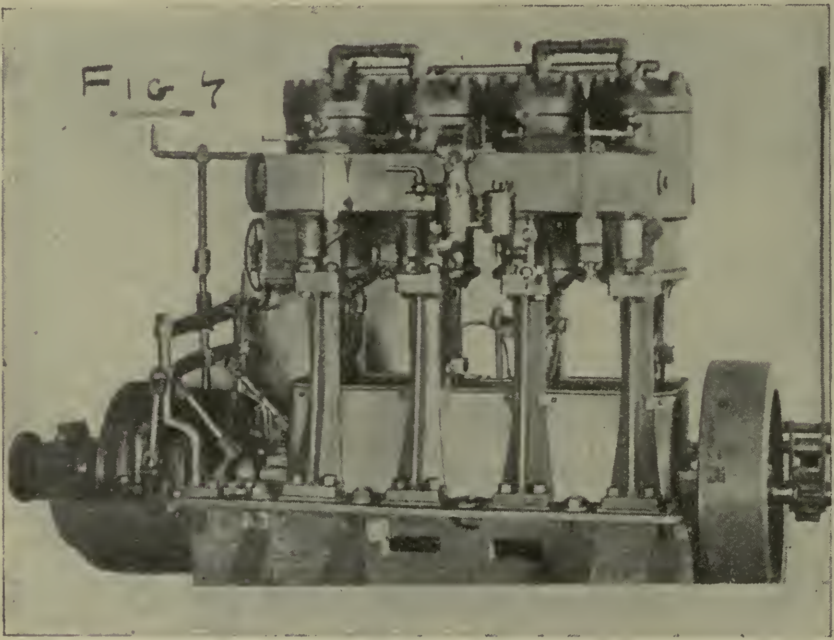


Fig 7

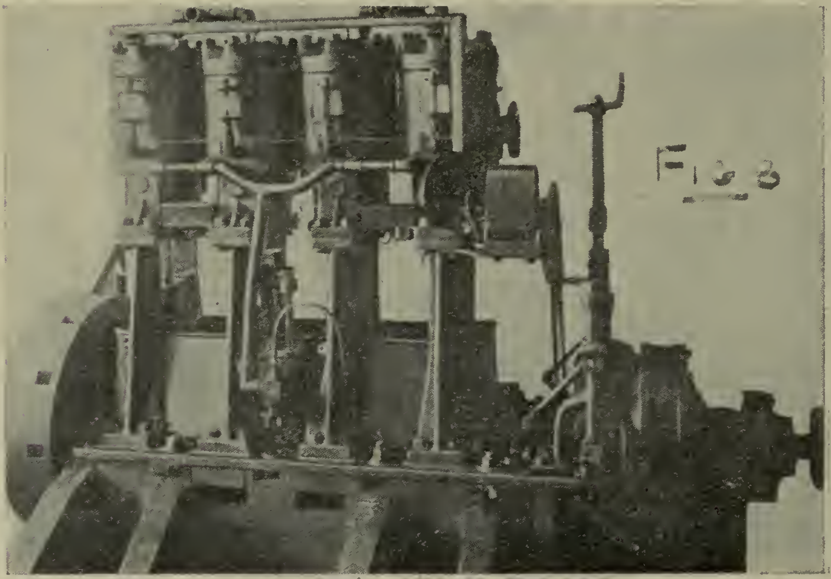


Fig 8

pattern in London: these engines were afterwards made on the Thames, but eventually the manufacture was dropped. The weak point of this engine is the short connecting rod.

The *Motor Boat* said of this engine at the time:—

“ The lubrication troubles hitherto experienced in two-stroke motors are eliminated, for since the charge never comes near the oil in the crankcase, none can be carried up behind the piston. Mechanically, the advantages are that the piston may have an easier fit; rocking and gudgeon pin troubles are eliminated; and there is no possibility of the cylinders being worn oval.”

The first open crankcase hot-bulb engine that I have been able to trace as actually on the market as a marine engine is the Bolnes. Fig. 9 shows this engine as first made in 1912: as can be seen, it has double columns. The later types have all been made with cast back columns and turned front columns.

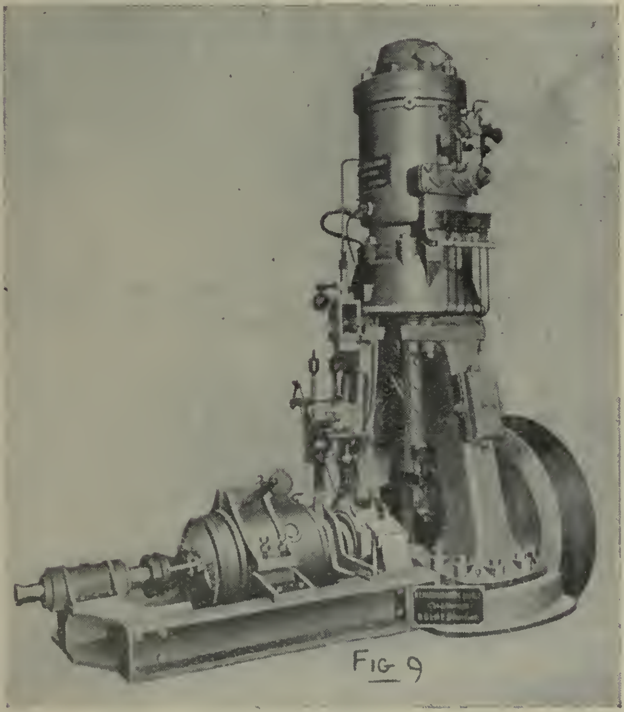


FIG 9

Figure 10 shows the largest power made up to the present date, 130 B.H.P. There is at present being built a four-cylinder engine of 320 B.H.P., and it is intended to commence the construction of a single cylinder engine of 150 B.H.P. These engines are coming largely into use: in the illustration the guards which are fitted have been removed to show the various working parts of the engine. As can be seen, it is made on regular marine engine lines, the pumps being worked from the cross-heads by levers as in steam practice. These will be seen better by referring to Fig. 21.

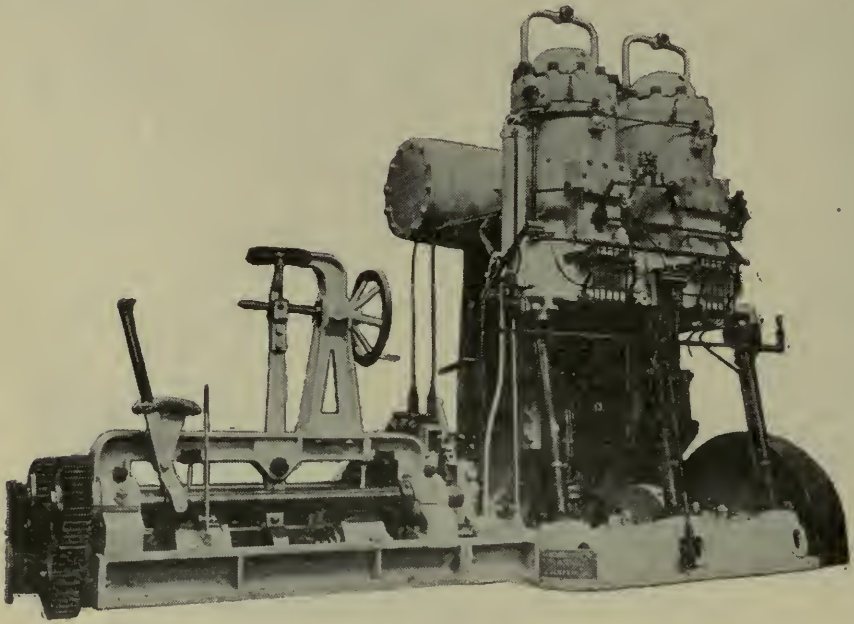
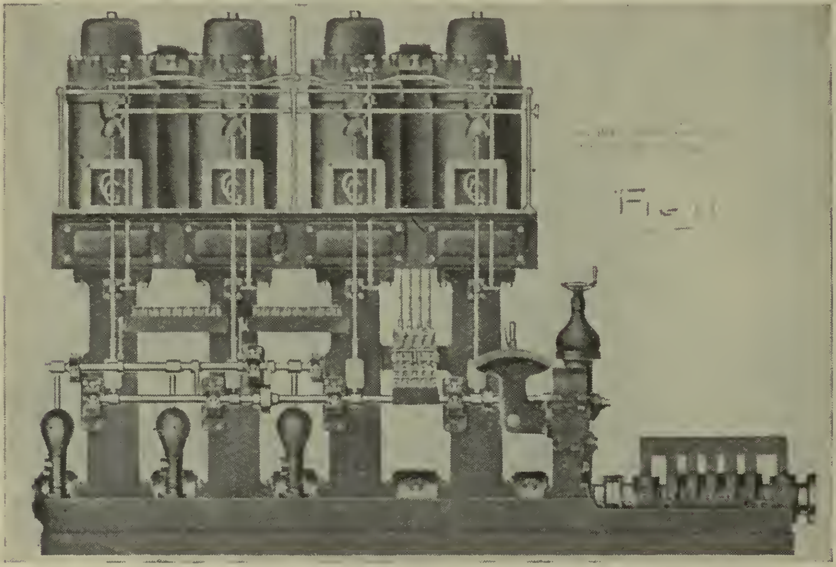


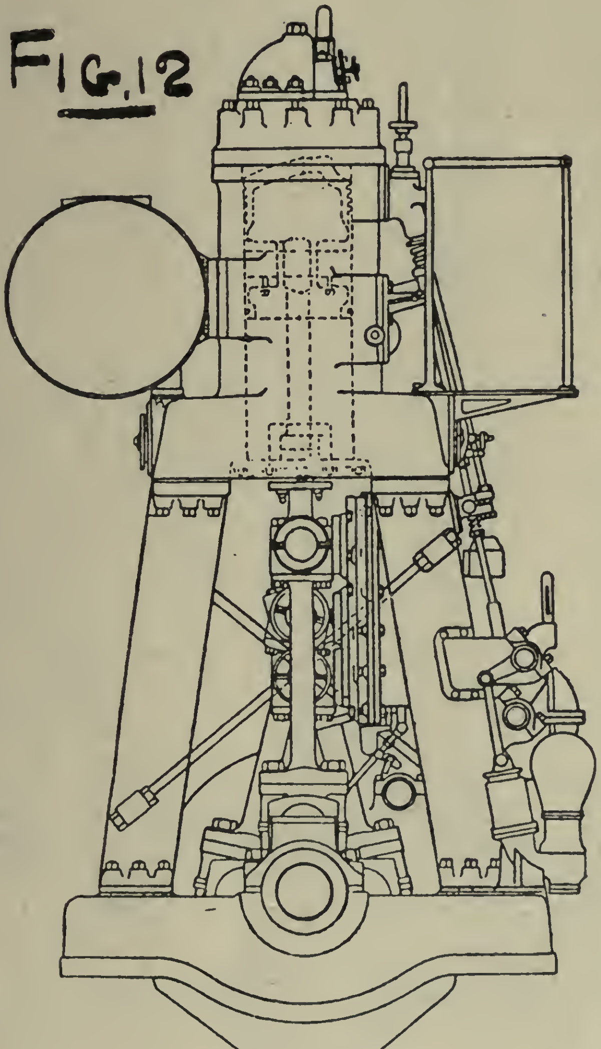
Fig. 10.

Figs. 11-12.—This shows the Grei engine, as made by Messrs. Gulowsen, Christiania, who made up to 360 B.H.P. in the



closed type. As can be seen, this is made with double cast columns, the bottom of the piston compressing the scavenging air in a very similar manner to the "Bolnes." The pumps are driven by cam-shafts in connection with spur gearing from the crank shaft: the reversing is by compressed air, the fuel pumps being driven by the cam-shaft which is fitted with a double set of cams—one set for ahead and one for astern. Circulating and bilge pumps are also driven off the main shaft. I am not aware if any of these have been actually built.

FIG. 12



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Figs. 13, 14 and 15.—This engine was designed by Messrs. Lambert and Adie, primarily to work on the Brons System of

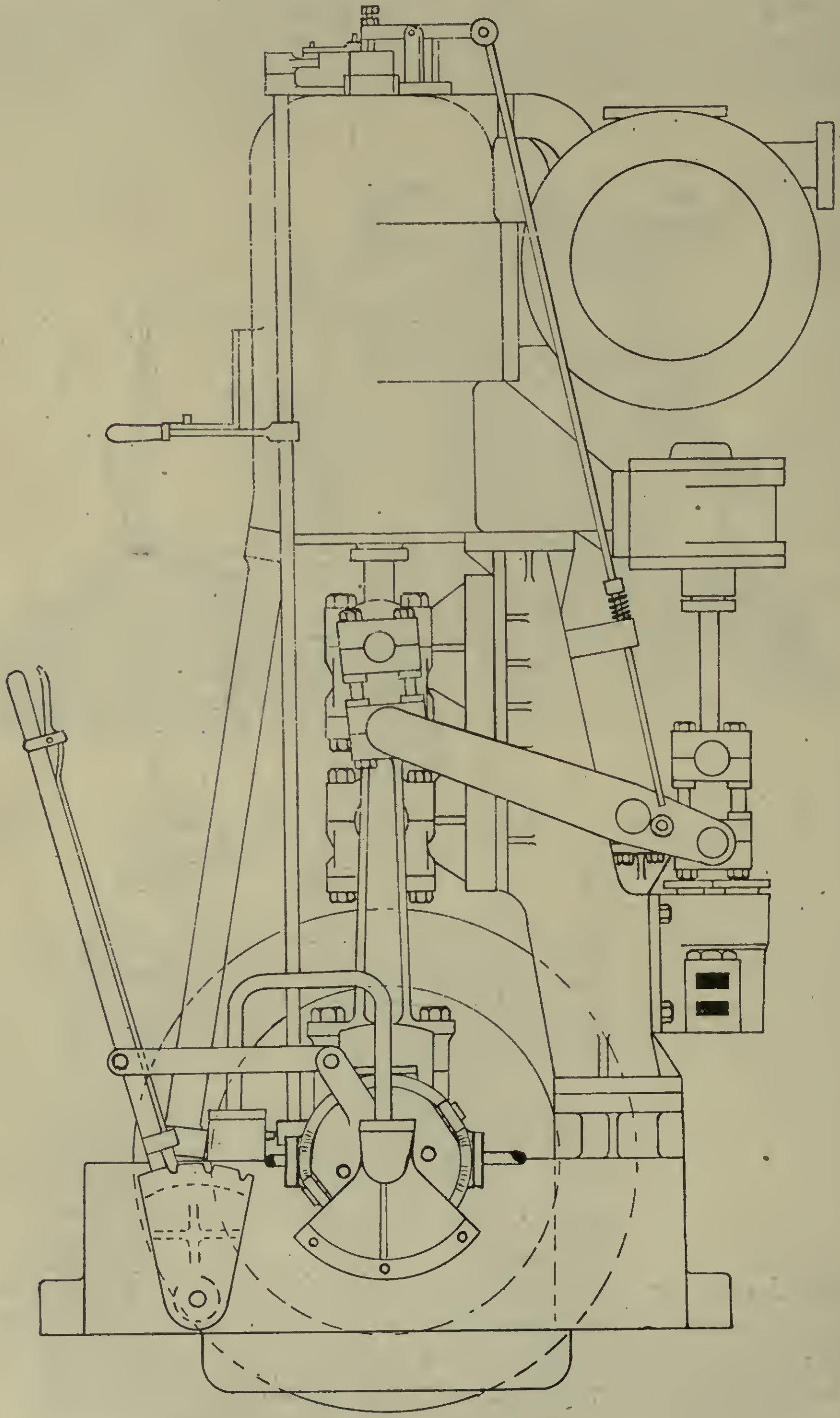


Fig. 13.

ignition, but can be adapted to surface ignition. In addition to the compression of the scavenging air under the piston, an auxiliary compressor for scavenging is fitted; this is about 20 per cent. of the capacity of the main cylinder. The pumps are driven by levers, as shown: forced lubrication is fitted: the main bearings and guides are also water-cooled, as can be seen by the

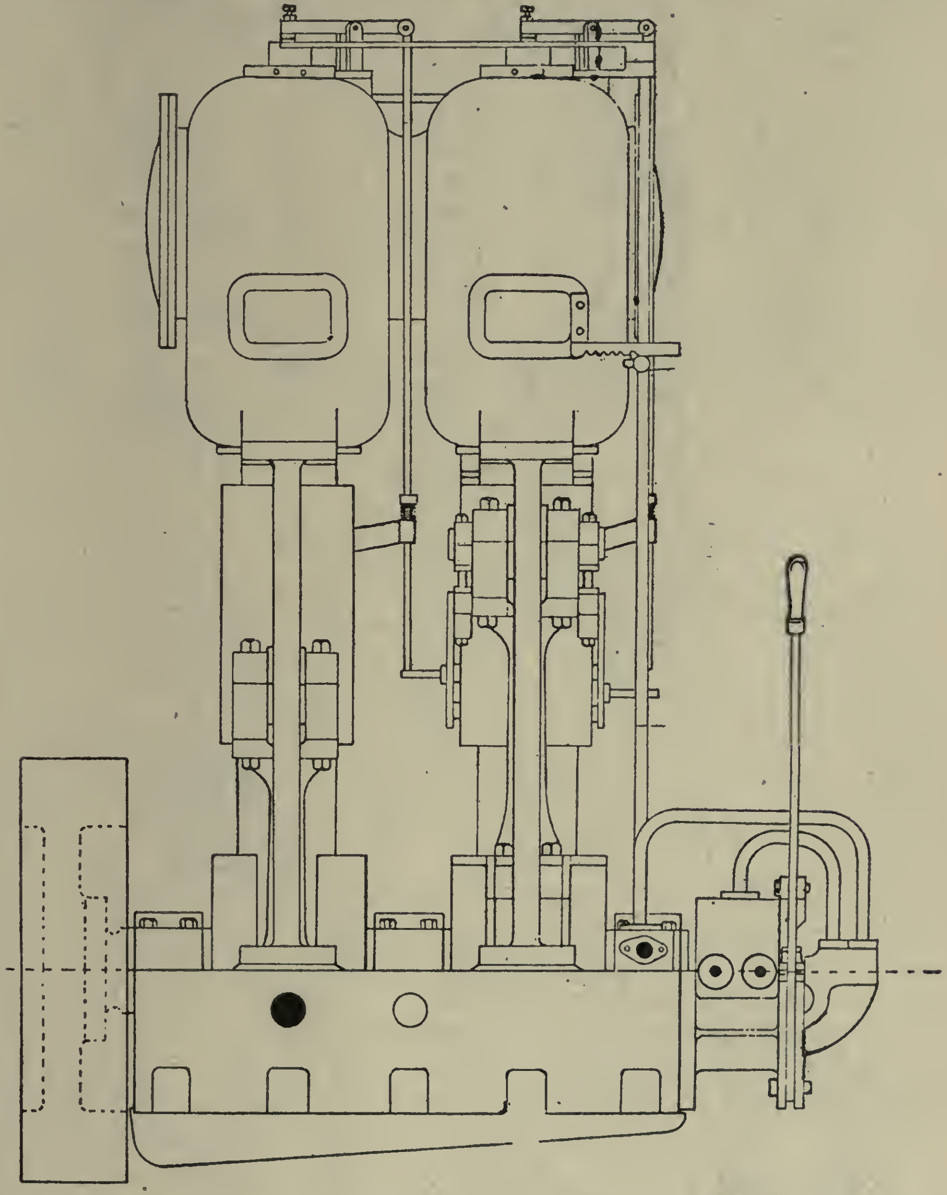
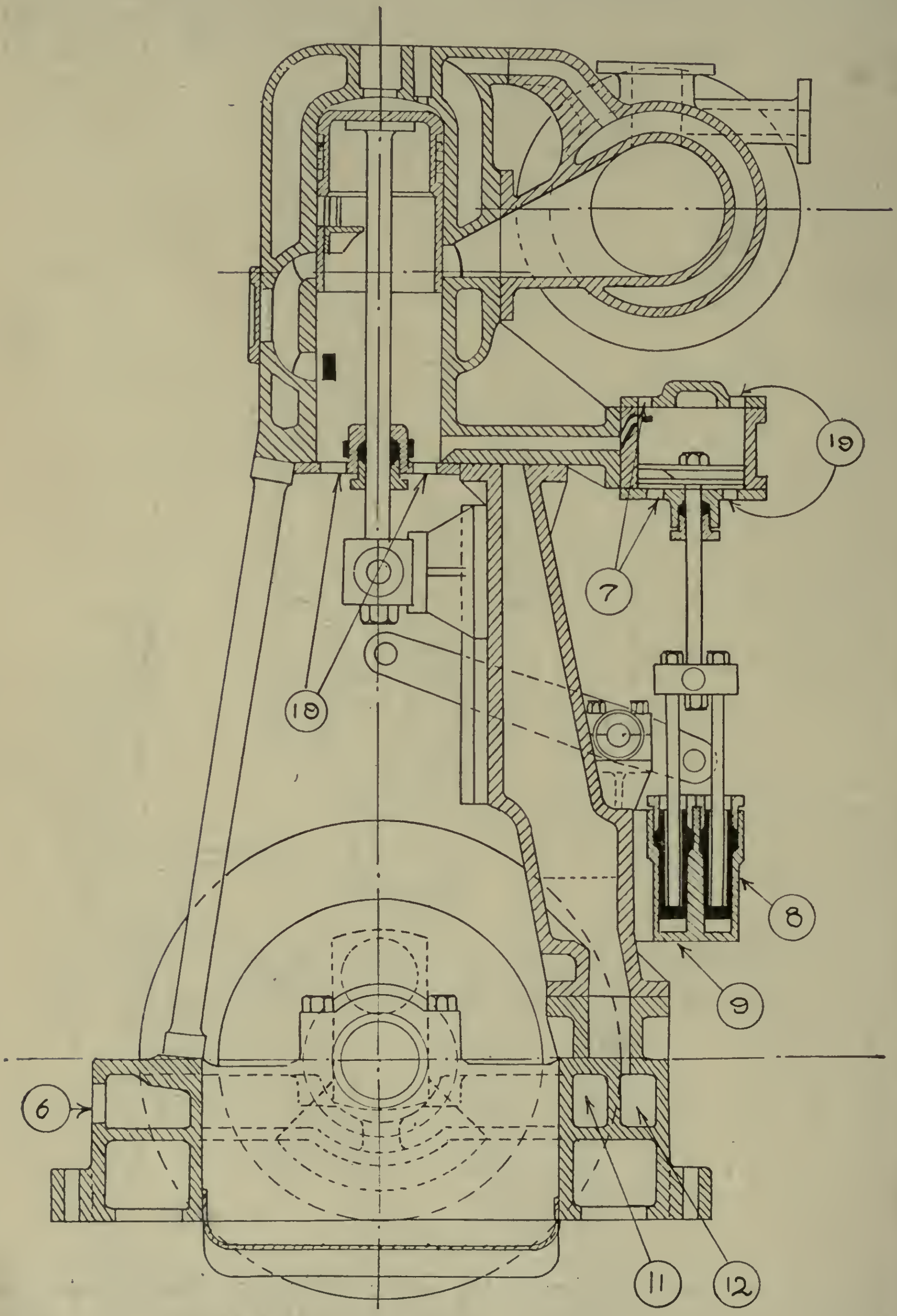


Fig. 14.

THE OPEN-FRONTED SURFACE IGNITION ENGINE.



Fig, 15.

sectional view. This engine has not yet been put on the market, but, to my mind, it represents a really serious attempt at simplification of oil engine design, and is well worth taking up by engine builders. Messrs. Lambert and Adie, like myself, were both in charge of oil engine repairs in France, and from our experience of some dozen different makes of engine, we came to the conclusion that the closed crankcase has many defects which would be eliminated by following steam engine practice.

Figs. 16 and 17 show a double acting hot bulb engine that has been recently patented: this was illustrated and described in a recent number of the *Motor Boat,* the publishers of which kindly gave me permission to use the blocks.

Personally, I cannot help thinking that the patentees have rather gone out of their way to introduce complications, and my comparison as to the relative cost of manufacture of closed engine and open engines would not apply.

I have not sufficient time to give a full description of this engine, but briefly, it is as follows:—

In a two-cylinder set, as shown, there is one working cylinder, and two scavenging cylinders. Referring to the Figure which shows Cross-section: A is the power cylinder, whilst B and C act as scavenging and induction pumps: a pin connecting the three pistons passes through the walls of the power cylinder working in the slots provided: an arrangement is fitted for sealing this, consisting of the side rods E. and F. Fig. 17. It is claimed that this arrangement is effectual, but I must say I think it is rather doubtful. The working cycle of this engine is as follows: On the downward stroke of the engine, air is drawn in through a valve into the cylinders B and C: on the upward stroke, the air is trapped after two-thirds of the stroke is completed; the passage K. Fig. 16 communicates with the top of the pump cylinders B and C and the bottom of the working cylinder A, where, upon the downward stroke of the engine, compression takes place. At the proper time the explosion takes place. Whilst this has been going on in the bottom of the cylinder, a similar cycle of operation is going on in the top end. All the pistons and side rods are water-cooled. Circulating pump, fuel, lubricating and bilge pumps are driven from a rocking shaft operated by eccentric on main shaft; the air compressor drives off the same shaft. The engine shown is rated at 60-70 B.H.P. at 500 revs., which is a very high speed for the H.P.; space occupied is said to be 5 ft. x 2 ft. x 4 ft.; weight approximately

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2 tons 5 cwt. Reversing is by air; the compressor allows supplies of air for injection with the fuel. There are two points against this engine in its present form; in my opinion, the sealing arrangement to cylinder A, and all the pumps centred on the one eccentric.

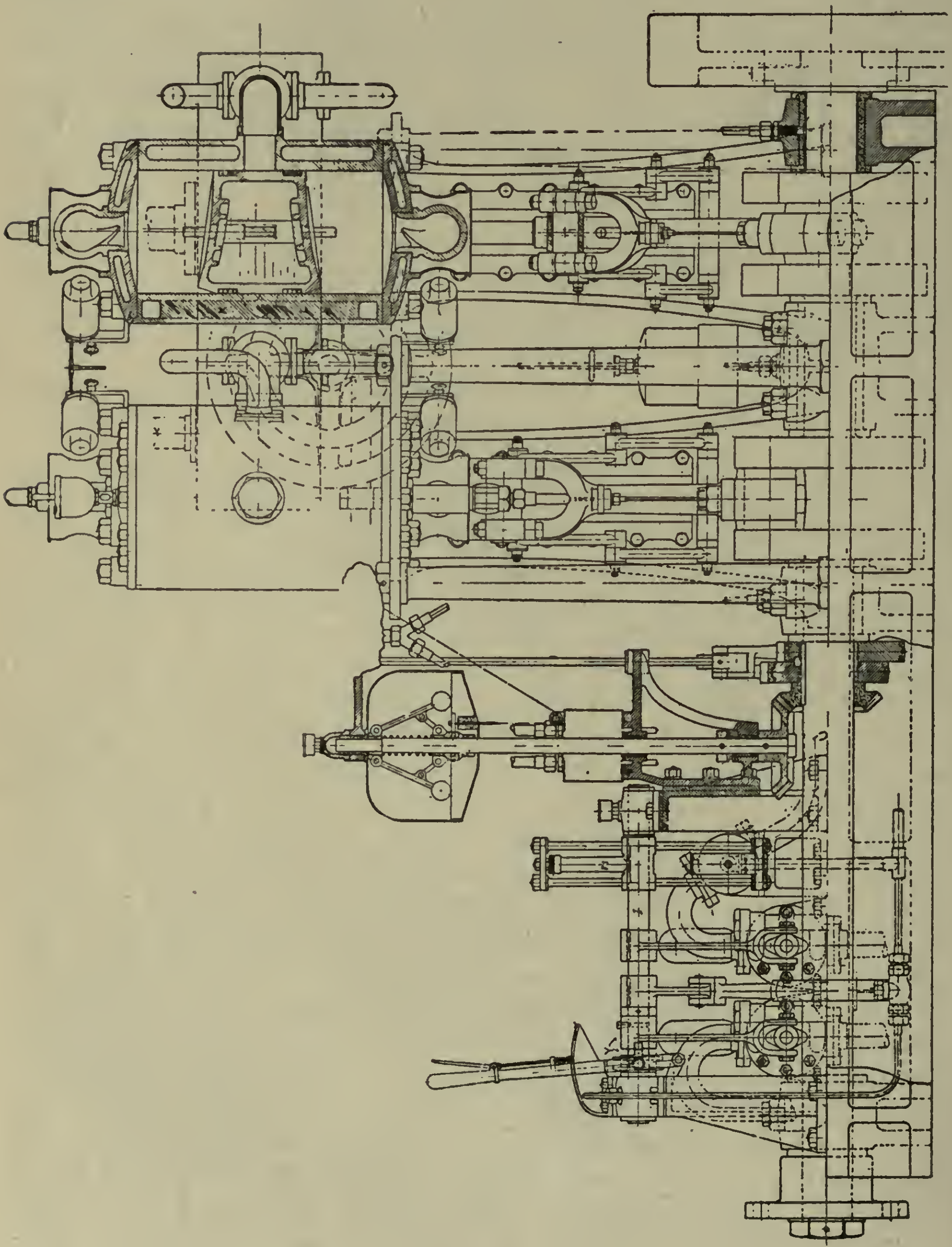

Fig. 16.

THE OPEN-FRONTED SURFACE IGNITION ENGINE.

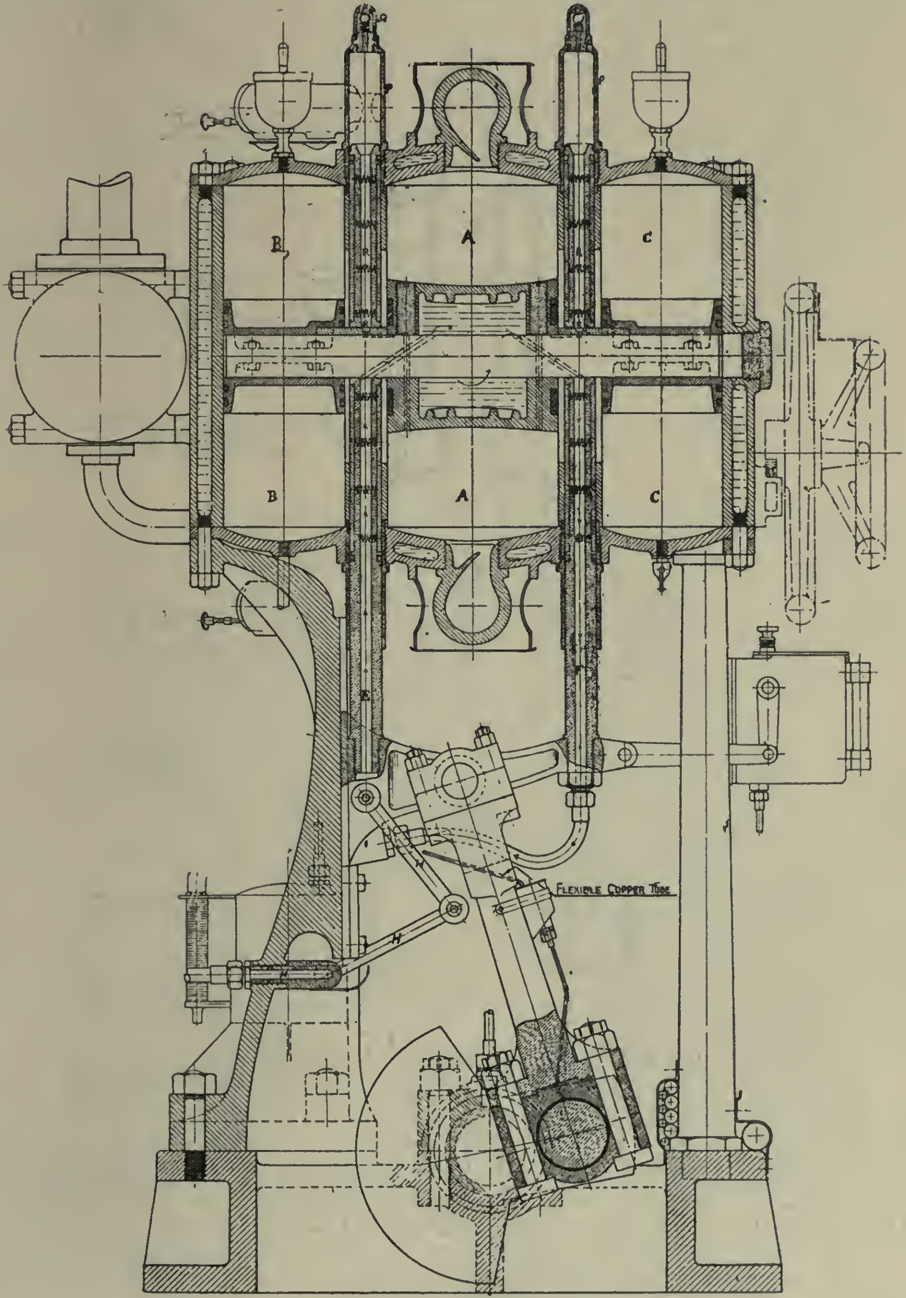

Fig. 17.

Fig. 18 shows another type of double acting hot bulb engine, the Stallard. As will be seen from the sectional view, the bottom hot bulb is placed at the back of the engine. A separate scavenge pump is fitted, worked as shown, by levers; this pump is designed to give a pressure of about 80 lbs. per sq. inch. The piston and rod are water cooled. I am not aware if one of these engines has been actually built.

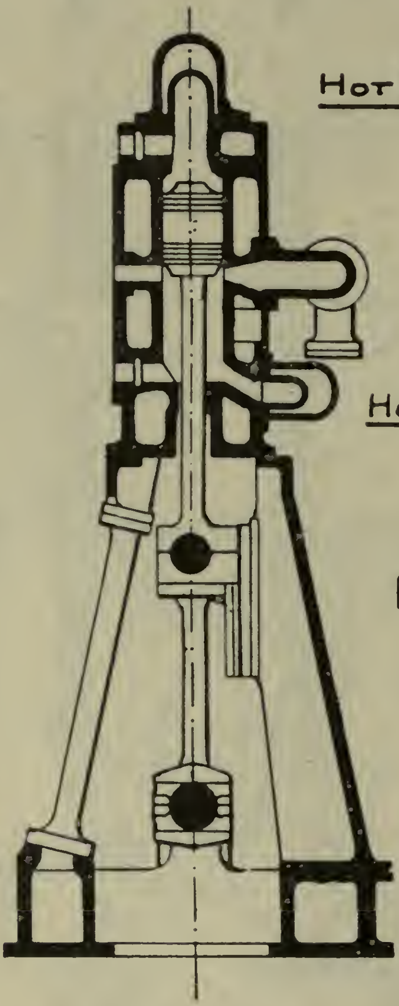

Fig. 18

— Stallard Double-Acting Engine —

I hope I have not made myself wearisome, and I am afraid that I have only been able to touch on the principal points, but if I have been able to show a little of what has been done to do away with what is one of the principal objections of the marine engineer to the modern marine hot bulb engine, I shall not be dissatisfied. I think I may add also that the engineer in charge of Diesel or Semi-Diesel engines at sea and on shore would sooner have an engine where he would be able to see and get at more of the working parts than is possible at present.

In conclusion, I should like to have shown some more of the earlier attempts at open engines, but it is not easy to get particulars of these without a great deal of research for which I am afraid I cannot find the time. I must also thank Mr. Walter Pollock for his permission to publish the illustration of the Stallard engine; and Messrs. Lambert and Adie, Anderson, Lewis and Siddal, H. W. Sumner and Co., and the editor of the *Motor Boat*, Mr. Chalkley, for all the information which they have supplied me with.

Figs. 19-20 represent the Sumner Engine. This represents the largest engine of the hot bulb or surface ignition type made

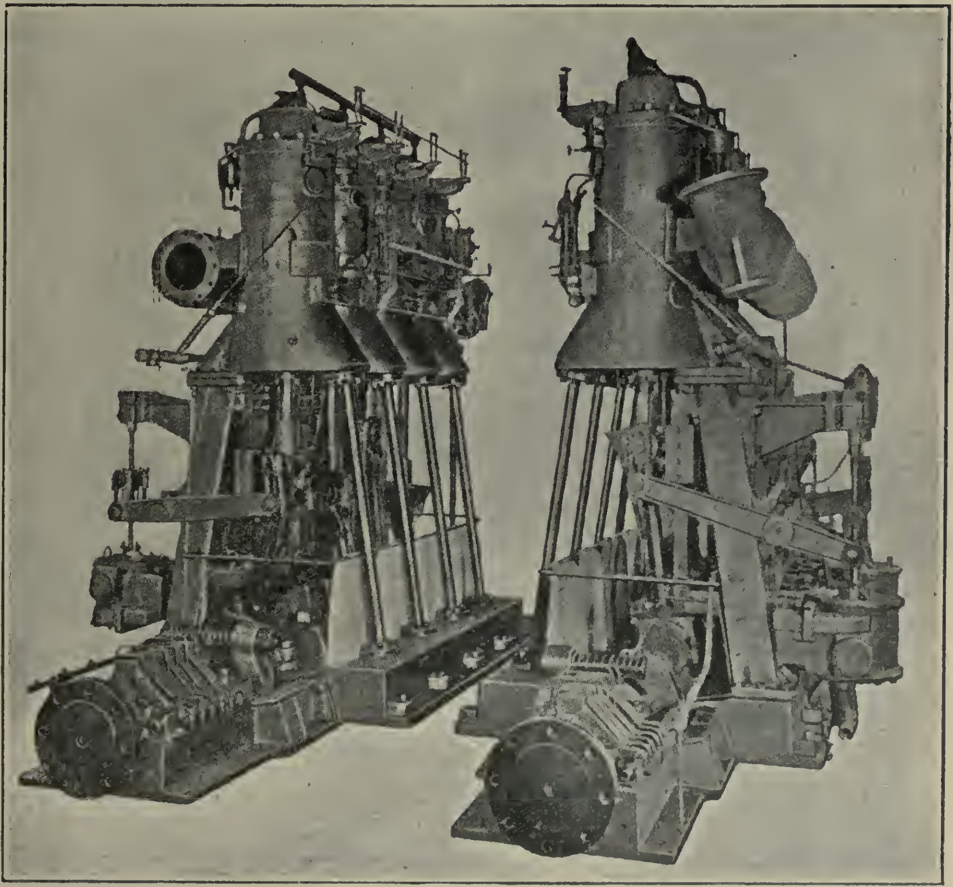

Fig. 19.

400 S.H.P. H. W. Sumner Company Marine Heavy Oil Engines.

THE OPEN-FRONTED SURFACE IGNITION ENGINE.

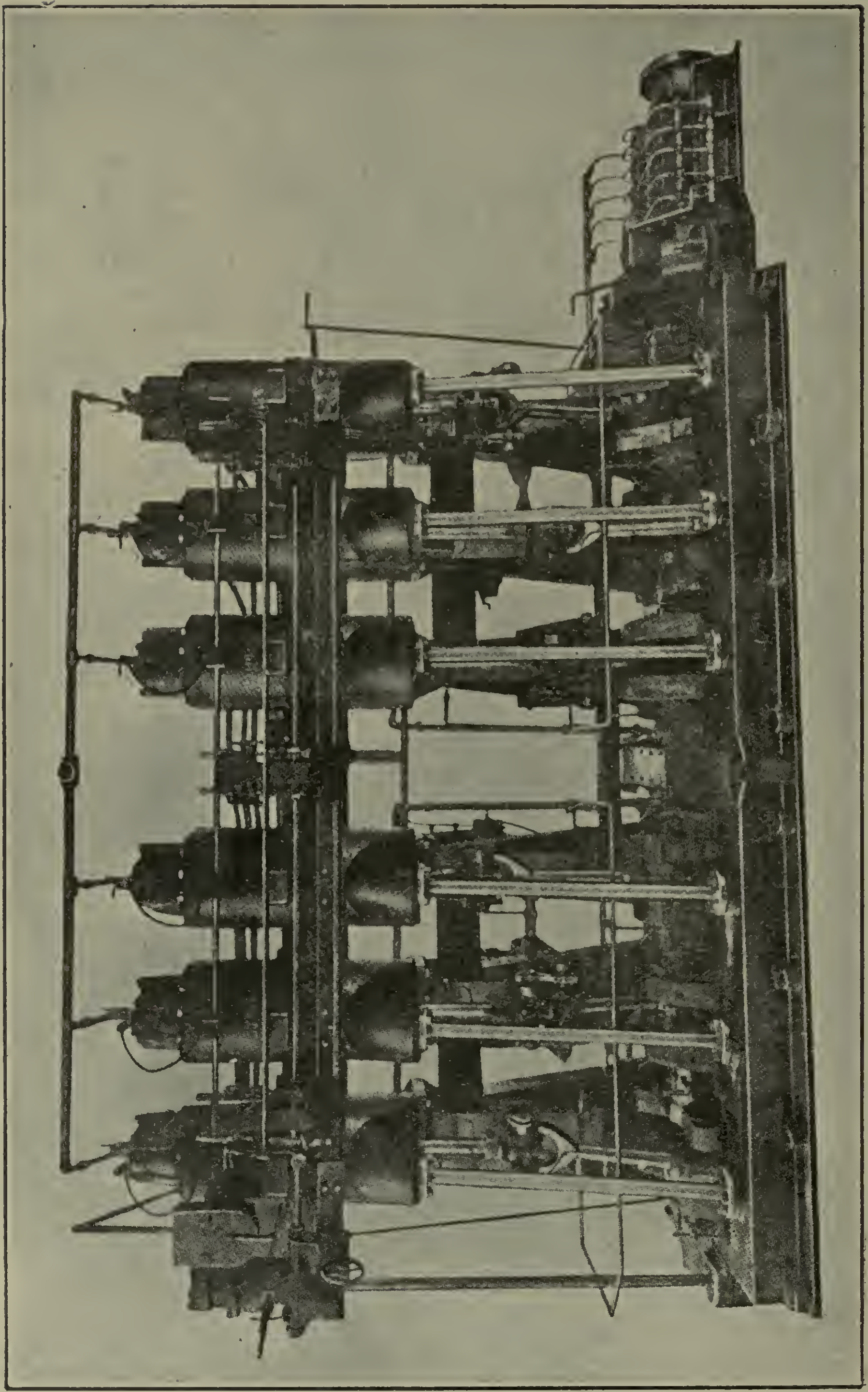

Fig. 20.—Front View. H. W. Sumner Company, 600 S.H.P. Marine Heavy Oil Engine.

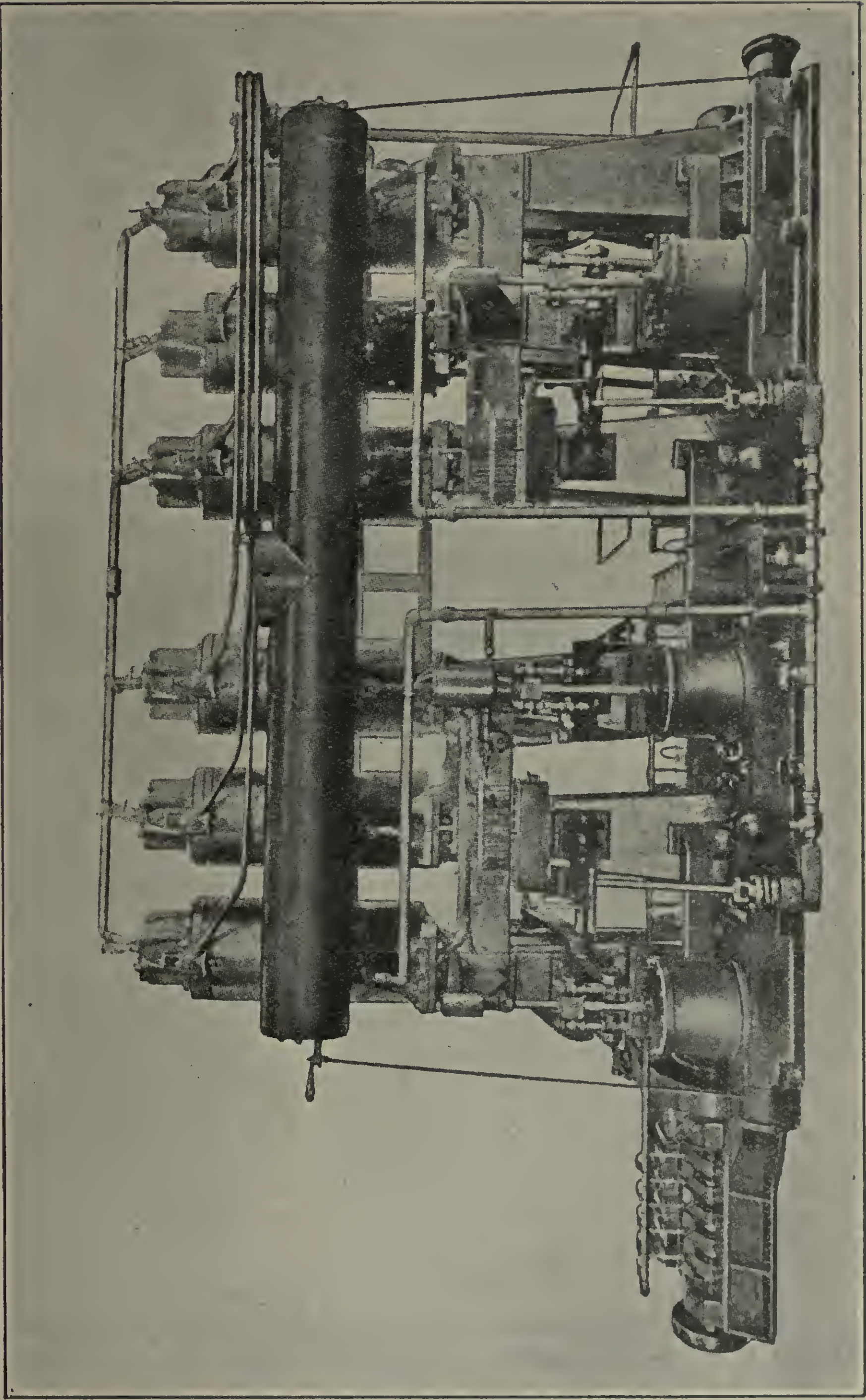

Fig. 20.—Back View. H. W. Sumner Company, 600 S.H.P. Marine Heavy Oil Engine.

as an open-fronted engine. The builders of this engine took a very bold step, as instead of building in the small sizes, one of their first engines was 600 H.P. It possesses many notable features: electric ignition, forced lubrication and separate compressors for scavenging air: it has, I believe, proved very successful, and I am sorry that I have never had the opportunity of seeing one at work. The other illustrations represent engines that have been proposed, but I have no knowledge if they have been actually built and tried commercially.

In order to give a better view of the" Bolnes " engine Fig. 21 is an end elevation in section.

Since writing this paper I have had the opportunity of reading extracts from the report of Mr. R. E. Mathot, one of the best known continental authorities on oil engines.

He has been studying the question of the use of vegetable oils for use with various types of oil engines, made by English, Italian, Belgian and Dutch engineers.

He reports very strongly in favour of the open type of engine. A very full report of Mr. Mathot's investigations will be found in *Gas and Oil Power*, May 5th, 1921.

A Member: The author speaks of the piston rod being flanged to the piston and at the other end tapered to go into the crosshead; should it not be tapered at both ends for nutting-up at each end? I think all marine practice and land practice is to have a piston rod nutted on to the piston, even if there is a spigot fitted on to it. In regard to lubrication, I agree with the author that the open engine has the advantage of being lubricated with economy. There is no question of the lubricating oil mixing with the internal combustion oil. That brings up the question of cross-head pins. I have never seen a trunk built up engine, either steam or otherwise, that did not give serious trouble with the pins, even when case-hardened. These pins and brasses certainly will wear oval and not being able to let them together, the swing of connecting rod partly inside a trunk piston does not lend itself to the best work, and if they are solid brass it means a new brass and pin, or if it is a split brass the thickness taken off the pin wants the clearance taken off also. With the four-cycle engine, at one time the engine is pressing either on the compression or explosion, but there is another time where the engine is light and the flywheel is running away with the piston rod, and therefore you have not an even bearing all the time.

A Member: I am rather surprised to hear the previous speaker say he does not know what a piston rod bolted with a T-end to a piston is. There are any amount of them in gas engines.

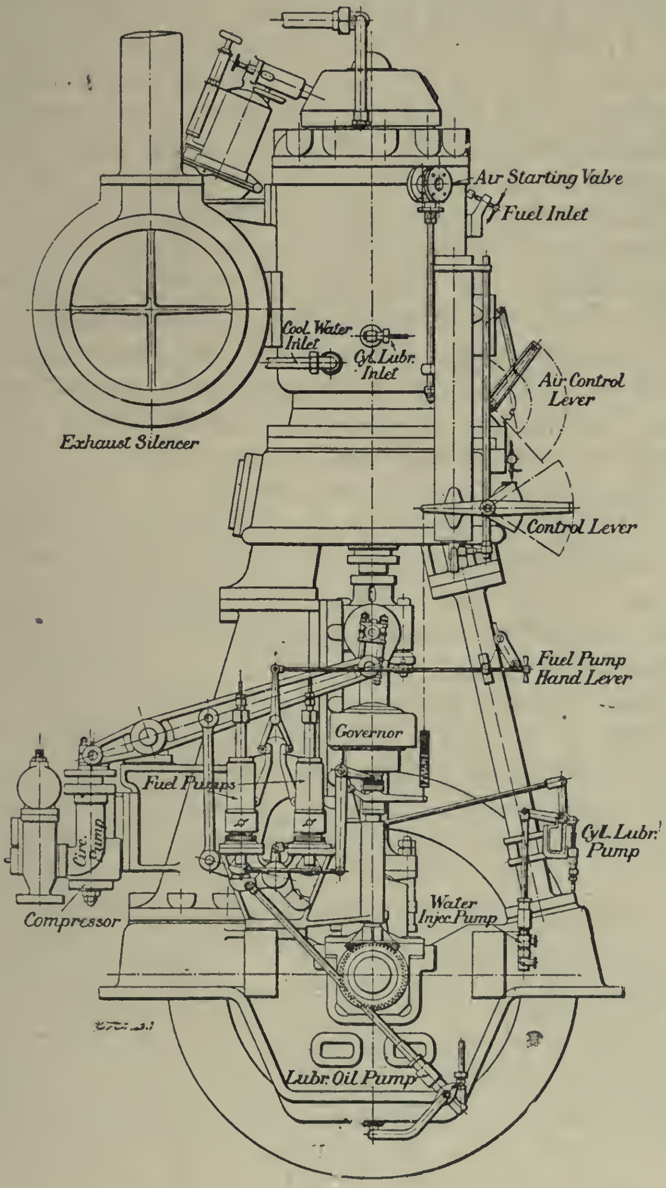

Fig. 21.

There is no necessity, when you have a constant thrust, to have the usual marine fitted piston; when the greater thrust is all downwards you can have a T-ended piston rod bolted to the piston sufficiently strong to carry any strains from the scavenging side. Further, in regard to the gudgeon pins: We have any amount of gas engines fitted with gudgeon pins and trunks, and we have very little trouble with them,—very little trouble indeed. The great point is surface. Very great trouble, comes when trying to undercut the amount of surface that is necessary for satisfactory working. If the surface is of ample area I think you will find that a good steel gudgeon pin with brasses will not give trouble. Not that I am in favour of that class of connection, I only refer to it because the previous speaker referred to it as being troublesome. We have been given the idea that it was almost a necessity for the oil engine to have it enclosed. No doubt, after years of experience with an engine in which you practically can see and handle all the moving parts, that was a point that went very much against marine engineers, as a body, taking to their bosoms readily the oil engine. The author has shown that it is not a necessity. I don't think a marine engineer cares whether he uses gunpowder, or oil, or steam, as his motive force is in the cylinder; but he does like to be assured that what is going up and down, and round, is not trusting to Providence for its success.

The Chairman: The author states in the paper:—"The use of trunk pistons which is necessary with a closed engine, is responsible for a lot of this wear." Why is it necessary to have a trunk piston with an enclosed engine? I have seen closed engines of the Diesel type with an ordinary piston, only deeper, and the piston rod instead of going through the piston was connected by means of a flange and studs in the bottom of the piston, the crosshead guides and connecting rod were in every way the same as an ordinary steam engine, this arrangement has proved quite satisfactory, and the top and bottom ends and guides are easily overhauled. The only reason I can see for a trunk piston is a saving of head room. With regard to gudgeon pins being fitted with split bushes and a tap bolt through the top of the connecting rod to adjust the brass, this did not prove to be satisfactory, and the solid bush, fitted tight into the connecting rod and a nice working fit on the gudgeon pin had to be adopted, with the latter practice it is necessary to make sure the pin is not too tight, and to see that the oil is getting to the pin before the engine is started, otherwise the engine will pull up after making a few revolutions, then you will have all the trouble of lifting the cylinder head, and taking out the piston and connecting rod to free the pin before making another start.

Regarding lubrication, my opinion is that forced is the best for high speed small engines, but with large engines I fail to see the advantage, because if you have forced lubrication it is necessary to have a closed engine, which is more costly to build, and the engineer in charge can't tell if every bearing is getting the correct supply of oil.

A Member: I congratulate the author on his advocacy of the simplification of the oil engine. But I do not think there is anything against enclosed engines. We cannot, I think, trust the piston to be the guide and I can see difficulties in using the crank case for compression. I have in my mind the electric light engines of large power made by such firms as Belliss and Morcom and Willans. Those engines are absolutely reliable. I think I can challenge any engineer to suggest any open type engine which will give the service those engines give. It is due mainly to excellent workmanship and careful selection of materials. It is due in the case of the Belliss engine to the successful system of forced lubrication. As a marine engineer who has watched both types of engines for a considerable number of years I cannot conceive of any system of drip lubrication doing so well. I would suggest that this matter be thrashed out by this meeting and at subsequent meetings in the endeavour to discuss the two points. I would try to maintain the closed trunk case and a satisfactory system of forced lubrication. The illustration showing the crosshead with split bush and set pins is to my mind a freak, and I think it ought to be set up as an illustration to be jeered at instead of one suggested by practical oil engineers. It is a thing that might be put on a toy, but not one to go to sea with.

A Member: I venture to state that what is required for marine engineers is an engine that will inspire confidence, and I feel quite sure with the enclosed engine this is far from doing so, and I maintain that when you enclose an engine and trust to forced lubrication, everything inside the casing is out of sight and must be largely left to conjecture until something goes wrong (such as a hot bearing, etc.), then it is too late to do anything to alter it.

It has been mentioned that the only way to find out if the bottom ends were running cool was to feel the two main bearings and when they became hot it denoted that the metal in the bottom end was probably run out.

I would ask any engineer if this is likely to give him any confidence in such an engine as he describes.

The last speaker mentioned some engines in electric light stations on shore, where engines with forced lubrications have run well. I have also seen the same thing, but they were generally small engines which depended very much on the oil they used and the attention they received, and if anything went wrong it was an easy job to 'phone to the makers for new spare parts or a new engine complete. I am afraid we must place ships in a different category to this.

We all know that ships have a nasty habit of rolling badly, also taking heavy lists to port or starboard and sometimes big seas have been known to come down the skylights.

It is quite apparent to every engineer that the internal combustion engine is going to depend very much upon the human element who will be responsible for the efficient working and upkeep.

We all know what steam has done in the past and is still able to do; I therefore think it will be a long time before the steam engine has passed away; as more oil is used, its price must increase, and then we shall have to come back to coal again.

I think that marine engineers to-day want an engine that is equal to the steam engine for reliability, and please let us have an open-fronted job so that we can see what is going on, also an efficient system of oil lubrication which shows at a glance where the oil is going to.

A Member: There is a point about this gudgeon pin which seems somewhat perplexing. The author make a very pointed remark upon the adjustment of this particular bearing, and that it is done cold, *i.e.*, when the engine is cold; and with him I very much agree that it is an unfair thing, no matter how fine the mechanism is, to him who adjusts it. There is the possibility of over-heating in the cylinder walls, so that his adjustment may be thrown all out, through it being adjusted cold. I think a fork connecting rod is far to be preferred to a single connecting rod. Even with a shorter bearing you get nearly twice the bearing surface circumferentially. There is less shock taken up by the connecting rod bolts when the crosshead is of large area. This is a moot point, but it is one that has given engineers a lot of trouble; that is, that connecting rod bolts should be annealed every year or every million revolutions, or should be renewed every year owing to the crystallization of the steel in the body of the bolt due to this explosive shock, anything from 40 to 120 per minute, varying with the speed.

The Author: As regards the question of the flanged piston rod, I think that has been answered for me. You will quite understand how it is fitted. Then in regard to forced lubrication. Well, I look at it this way. If people want to have forced lubrication they can have it, and if not, they can have the drip system. There is a lot to be said for both. Personally, as far as the Bolnes open-fronted engine is concerned I know the builders will fit forced lubrication if people want it, but up to the present it has not been fitted and the drip system has worked very satisfactorily. Our chairman has raised the question of the trunk piston. Of course, one of the points that the closed engine people make about their engine, is the small head room needed, and that is one of the things they generally argue against the open engine. There is no reason why a closed crank engine should not be fitted with a piston rod and connecting rod on the same principle of that first engine in this paper, the " Reliable," which has a trunk guide with a connecting rod. Of course the connecting rod should be made longer than shown to give good results. In respect of the remarks on the crosshead; as far as the Bolnes is concerned you can have either type of crosshead you like. I prefer the double crosshead. It is far easier for adjustment, and also easier for overhaul and gives better results.

A member writes as follows:—

After years of experience of open and closed type auxiliary steam engines on ships, his experience is that upkeep costs are much less with forced lubrication engines. It is reasonable to assume that they would be, because the supply of oil is regular and the enclosure of the engine is a safeguard against dirt.

In oil engines where the bottom of the cylinder is open to the crank chamber there is a liability of carbonized oil from the cylinders to mix with the lubricating oil, and it does mix. This has to be closely watched and when the lubricating oil becomes thick or heavily discoloured, the charge should be changed as oil is cheaper than metal.

Where there is a crosshead, the remedy is to fit a distance piece between the cylinder and the crank chamber. This was done on engines, the writer has knowledge of, with marked economy of lubricating oil afterwards.

With a Trunk Piston the above remedy cannot well be applied and the only remaining remedy is to catch in trays as much of the carbonised oil as possible before it drops into the lubricating oil.

The writer's opinion is that cylinders should be apart and divided from crank chambers and the working parts inside the encased crank chamber should be force lubricated because with any other system they would not get equally well supplied with oil.

SESSION 1921-22.

President: SIR JOSEPH P. MACLAY, BART.

The Marine Diesel Engine: Its Reliability in Service.

BY MR. ANDREW J. BROWN.

Read Tuesday, April 26, 1921.

THE intention of this Paper is to give the question of the Marine Diesel engine practical consideration and to try to show how its reliability is ensured.

Considering the cycle of operations; with the Diesel engine of to-day we have two types to choose from, viz., two and four stroke. This has been the subject of controversy ever since the advent of the Diesel engine, the leading authorities being divided in their opinion, some advocating the two and others the four stroke cycle. The relative advantages and disadvantages of both have been so often enumerated that it is scarcely necessary to investigate them again. The possibilities of simplification of the two stroke engine are great, where the cylinder head is concerned, but it is done at the sacrifice of the most vital part of the engine, viz., the cylinder liner. The presence of the exhaust ports in the cylinder forms the limitation of the engine, as it is through the bars of these ports that the most serious heat stress is set up. Where scavenging ports are used—and these must be adopted if the cylinder head is to be kept simple—the unequal temperature of the liner around the ports causes distortion and consequently affects the piston rings, resulting in unequal wear throughout the liner. In the author's experience of two stroke engines, both port and valve scavenging, the great weakness and the unreliable part has always been the bars of the ports. If scavenging valves are resorted to, the cylinder head has no advantage in design over the four stroke, and judging from many existing designs of two stroke engines where three or four scavenging valves are used in addition to the fuel, starting, and relief valves, the design of the head is much worse than that of the four stroke engine where the number of valves is limited to five, viz., inlet, exhaust, starting, fuel and relief.

Perhaps the best way of showing the reliability of the Diesel engine is to investigate each point of design individually and consider what each feature entails.

BEDPLATES.

Examining current practice in bedplate design, it will be found that the general course followed is much the same as has been adopted in steam engine practice, viz., the cast iron box pattern which has been well tried out and is of proven reliability.

In one design of engine the bedplate sections are provided with male and female spigots which run all round the joints. The presence of the spigots prevents any variation in height of the main bearings, such as is sometimes experienced with bedplates in ships, due to the working of the hull in a seaway.

The general practice for main bearing bushes is to make the shells of cast iron or cast steel and line them with whitemetal. If a good brand of whitemetal is used and is properly secured to the shells, the bearings may be set up to 6/1000th clearance where forced lubrication is provided. Of course, good alignment must be ensured, while, if there is any doubt as to the alignment, such as caused through straining of the ship in a seaway or unequal cargo loading, it is better to keep the clearance at 10/1000th.

Main bearings show up extremely well under service conditions, the amount of wear being small. In the author's experience of four stroke, forced lubricated engines, there have been instances of 12 months running during which time the wear was practically negligible.

CRANKSHAFTS.

In the crankshaft of a big engine it is usual to divide the shaft into sections, as in steam practice, and build it similarly. In the case of a six cylinder engine, two sections usually suffice, the sections being joined by flanged couplings. The compressor crankshaft is in almost every case a separate section joined to the main shaft by either a flanged coupling or else by male and female castellations. In the Diesel engine crankshaft the space taken up by the couplings is not lost, as it is usual to mount the cam shaft driving gear on the rim of the coupling flanges.

In the slow running Diesel engine, which is the type under consideration, sufficiently good balancing is usually obtained by the crank dispositions, and so it is unusual to find counter-balanced crankshafts.

CYLINDERS.

In cylinder design great diversity exists as to type and construction.

In the four stroke engine the cylinder is a simple barrel whose function is to act as a water jacket for the liner, and, in most cases, to carry the tension load.

In the two stroke engine the cylinder is not so simple, as it has to provide an exhaust belt, and, in the case of a port scavenging engine, a scavenging air belt in addition.

The various designs of cylinders can be classified as follows :—

(1) Liner, jacket and cylinder head all one casting.

(2) Liner, jacket and cylinder head in two castings, the liner and jacket being cast together.

(3) Liner, jacket and cylinder head in two castings, the liner and cylinder head being cast together.

(4) Liner, jacket and cylinder head in three independent castings.

There are numerous examples of the above types to be found in service and from the results obtained, there is much to be said for each design. The advantages and disadvantages are more the concern of the designer rather than that of the marine engineer.

To give each class consideration from a practical point of view :—

Class (1) is generally confined to small engines and for inspection of the piston it is necessary to remove the whole cylinder.

Class (2). The liner and jacket being cast in one makes reboring somewhat difficult, and, in the case of a two stroke engine where cracked liners are not uncommon, it means greater expense in renewals.

Class (3). The liner and head being cast in one gives advantages in keeping the metal of even thickness throughout, and

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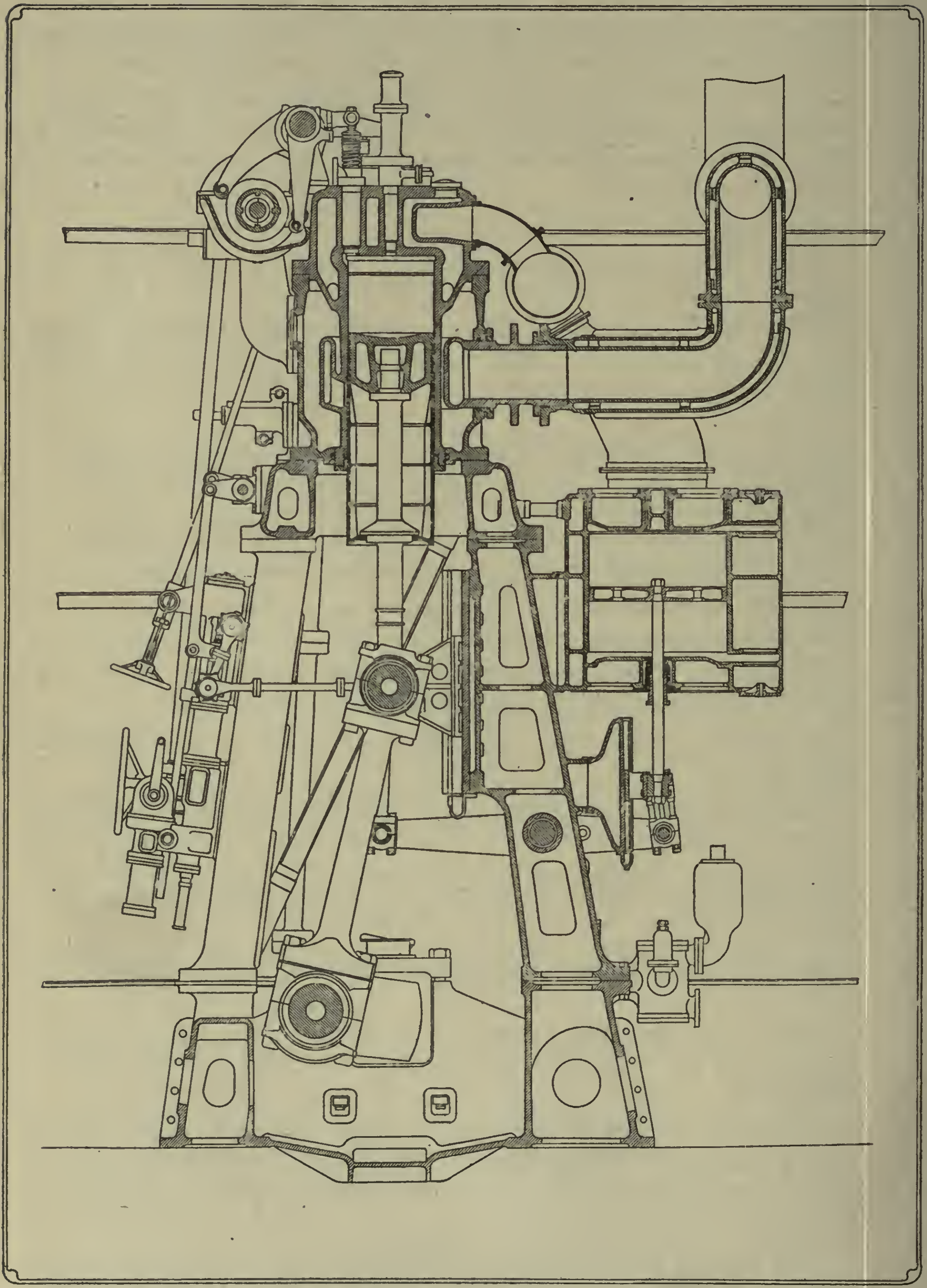

Fig. 1, Krupp Engine.

assists the cooling of the head, but it has distinct objections in that the two most susceptible parts of the cylinder are in one, and fracture of either means renewal of both. Moreover, inspection of the piston is rendered more difficult unless considerable ingenuity is displayed, as in the case of one Continental design of engine where the pistons are removed for inspection from the bottom of the cylinder.

Class (4) has probably found greatest favour amongst designers, and is the most attractive, as each part is separate and therefore of the simplest form.

In the mounting of the cylinders many means are adopted. Fig. 1 shows a section through one of the two stroke "Krupp" engines of M.S. *Hagen*. The cylinder of this engine comes under category (3) while it being a two stroke, valve scavenging engine, the exhaust belt will also be noticed. The cylinder is bolted down on to a cast iron entablature which is, in turn, supported by cast iron columns.

Fig. 2 shows a section through the well known "Burmeister and Wain" engine, from which it will be seen that the cylinder comes under category (4). The bottom of the water jacket forms a distance piece and is secured to the "A" framed crankcase by means of long bolts extending upwards from the bedplate. The cylinder head is secured by means of studs screwed into the top of the water jacket.

The "Beardmore-Tosi" engine cylinder (Fig. 1A) is also under category (4). The cylinder is bolted direct to "A" type columns, while the head is secured by bolts to the bulged water jacket. In this engine the cylinder liner is provided with a spiral rib cast integral on its outside diameter surrounding the combustion space. The provision of this rib allows of greater strength where it is most needed and also assists the cooling, as the water is carried over the whole surface at high velocity.

Fig. 3 shows the "Werkspoor" design of cylinder and framing from which it will be seen that the cylinder comes under category (3) while supporting is by means of large turned steel columns extending from the bedplate.

The most general practice in Diesel engine design has been to adopt guides of the single slipper pattern, in which the ahead side is usually water jacketed. In design and construction the guides and crosshead shoes follow that of orthodox steam engine practice.

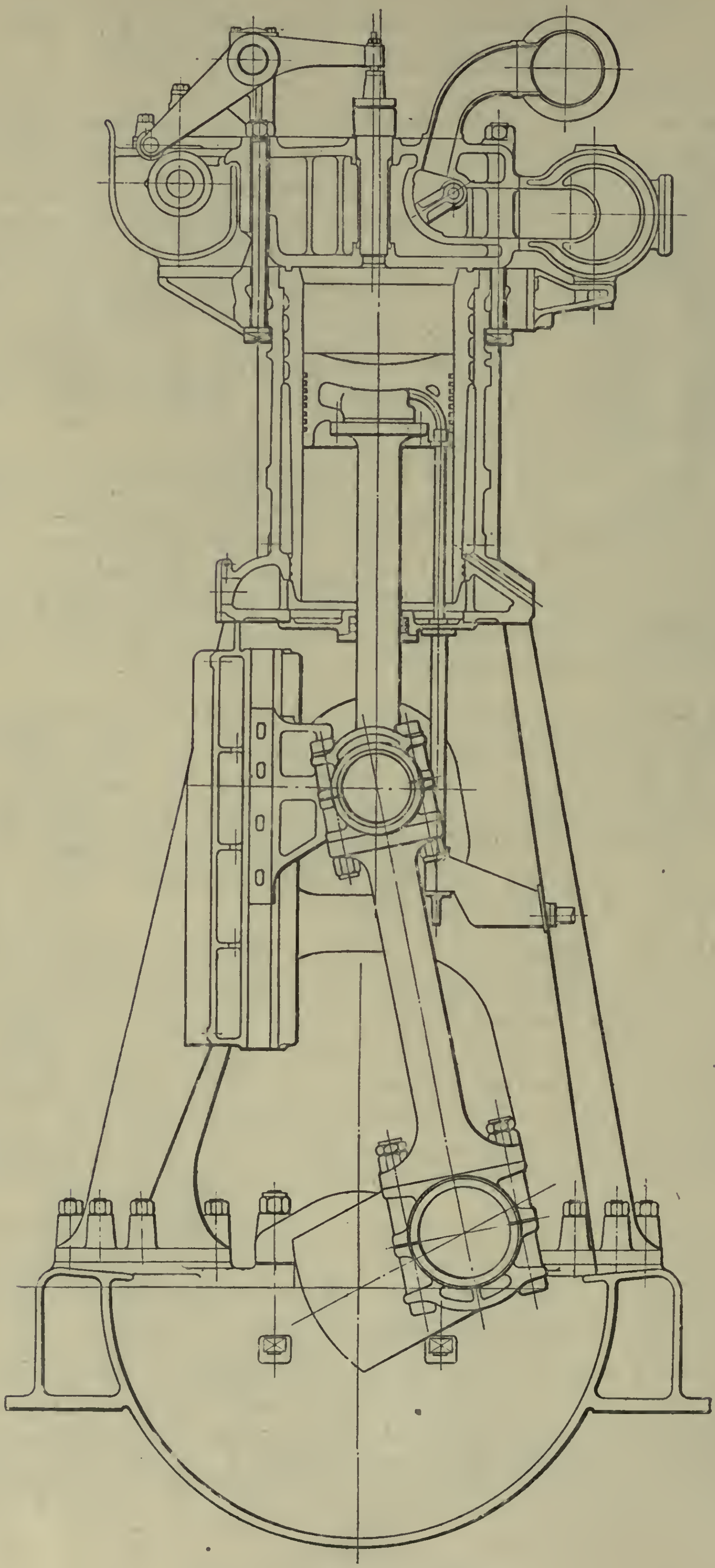

Fig. 1A. Beardmore-Tosi Engine.

As an example of what may be expected from good design and construction, the author's experience has been, that, with forced lubrication, a year's running without adjustment of the guides is quite normal behaviour.

Pistons and Piston Cooling.

In this connection, no standard has, as yet, been adopted. In all probability there are no two makers who adopt the same design of piston in all its features, although there are many points of resemblance in them all. In the majority of cases, the concave form of crown has been adopted, as this type allows of the easiest and most direct fuel injection being used. The convex form of piston top necessitates the use of radial injection, which is a drawback, while its greatest advantage lies in absolute safety in starting the engine. When an engine is being started and misfiring takes place, a pool of oil may collect in the piston top when of the concave form. When the oil does become ignited, if in the presence of starting air, a dangerous pressure results, which often lasts over a few strokes, while its effect on the engine is scarcely beneficial. A convex piston obviates this occurring, as any unburnt oil runs off the piston top. The effect of this is to gum up the piston rings and cause compression leakage, so it will be seen that the result of misfiring with any type of piston is highly undesirable, and, if anything, more so with the convex pattern.

Variation in the form of piston top is met with in several engines where a certain peculiarity has been adopted in order to meet special requirements.

Irregular shapes are undesirable, not only from the unequal heat stresses imposed, but from the point of view of repairs in the event of fracture. When the regular form is maintained, *i.e.*, convex or concave, a cracked top can be repaired by recessing the crown and inserting boiler plate, when the piston should be capable of considerable service before being scrapped. The author has had experience of this repair being carried out in several cases, and it proved, in every way, eminently satisfactory.

Fig. 4 shows a sketch of such a repair to an uncooled piston of 12½ in. diameter. It will be seen that the piston crown is recessed to the same curve as the original concave, the new diameter formed taking in the whole length of the crack. The plate inserted is made from a $\frac{7}{8}$ in. boiler plate machined and

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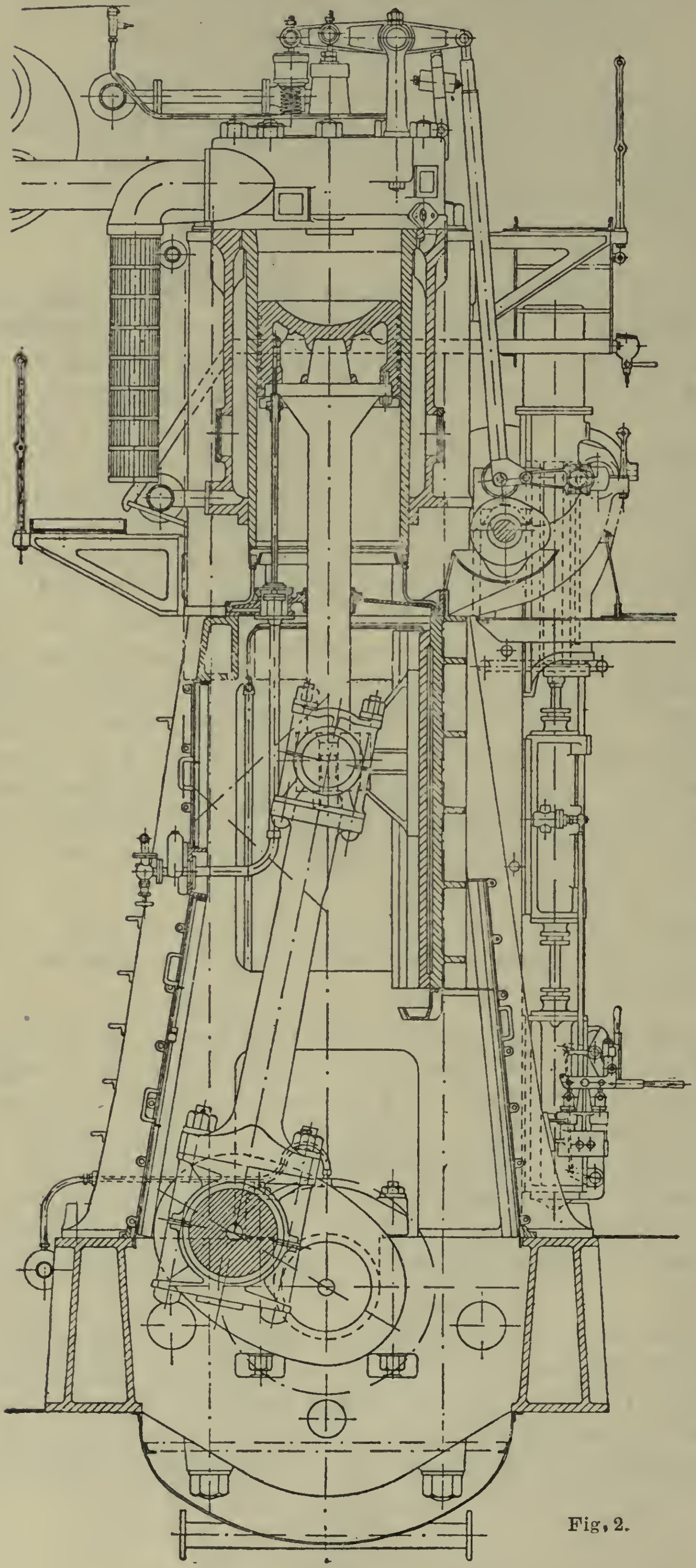

Fig. 2.

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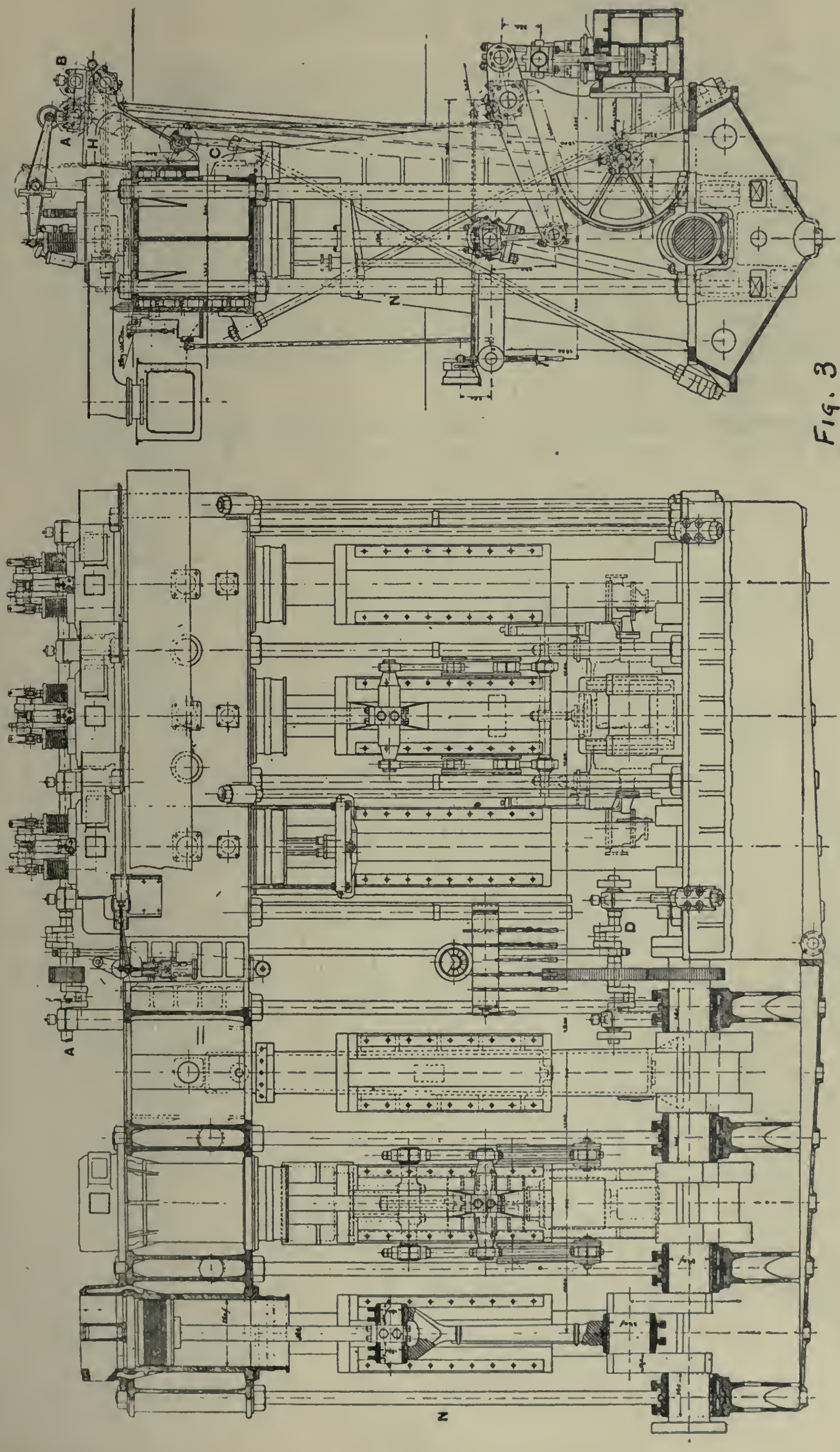

End View.

Front View.

Fig. 3

roughly bedded, and is secured by eight ½in. countersunk set pins. After securing the plate, the piston top is again machined to a smooth finish, when the joint lines of the plate and pin heads are indistinguishable.

In the author's case, after a year's running with such a repair, the joint lines of the plate and pins were only visible, and no slackness whatever was apparent: moreover, there were no signs of deterioration through combustion. It is the author's belief that such a repair is equally applicable to the largest water cooled pistons, provided the workmanship is good and the necessary expansion allowed for.

Regarding piston cooling, water is the medium most generally in use in nearly every large engine. Oil has been tried

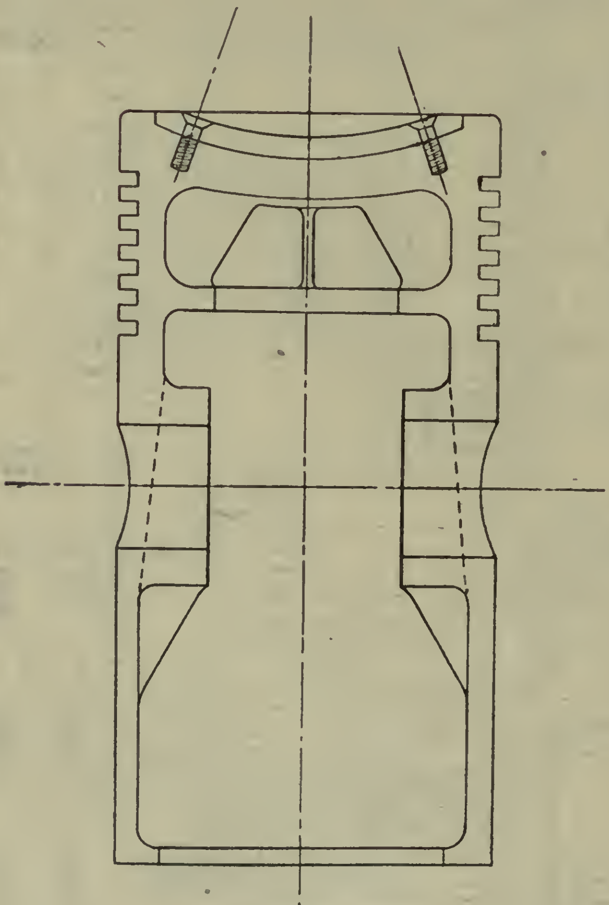

Fig 4.

in order to simplify design, but, unless the circumstances are exceptional, the results are not warranted. Due to deterioration of the oil through exposure to such great heat, the resultant wear in bearings is greater, while, of course, more oil is consumed than if merely used for lubrication, and its life is considerably shortened. Systems have been tried wherein separate oil is used for piston cooling, but as this entails the use of special gear, its advantages are mostly lost. Again, the design of an oil cooled piston calls for more care, as much better circulation must be ensured due to the greater temperature of the cooling medium and its lower specific heat.

Where water is the medium used for piston cooling, it is usual to find that fresh is preferred to salt. The use of fresh water entails additional complications in the way of tanks and coolers, but, all things considered, the system is amply justified. There is no appreciable loss of water through evaporation, and, with a good system, leakage is either very slight or else non-existent. With the quantity usually allowed, one filling of water will last three or four months, at the end of which time it is advisable to pump it overboard and clean and refill the system.

The importance of piston cooling water filters and the necessity for having them accessible may here be noted. The formation of scale and deposit is unavoidable in any water system, and its presence in rough weather becomes apparent through the filters choking and the pressure dropping. The only course open to the engineer on watch is to change over from the running pump to the spare and clean the filter that is choked. With the ship rolling and pitching, it should be readily appreciated that accessible filters are desirable, and yet, there are designs in existence where changing the filters necessitates going under the floorplates, and lying on one's back to the accompaniment of cooling draughts of bilge water with every movement of the ship. Such is an example of design, wherein the designer has apparently trusted to a special manifestation of Providence preventing the necessity of his gear ever needing examination.

The best design is that where two interchangeable filters are provided to each pump, or else that of having the one filter of each pump interchangeable with that of the other, and, most important, having them accessible. Filters are essential, as if any scale or deposit gets into the system, trouble may be encountered with overheated pistons and undue wear of parts.

Fig. 5 is a diagrammatic sketch of a piston cooling gear of the telescopic pipe pattern. This is the most satisfactory type of gear in service, and little, if any, trouble is experienced with it. The only glands incorporated are easily kept tight and seldom need repacking. The success of the gear lies in perfect alignment, and this must be ensured. Mal-alignment means that rapid wear will take place in the tubes, which become useless in consequence, as the wear, being all on one side, makes it impossible to cure the trouble by repacking the glands.

Fig. 6 is a diagrammatic sketch showing the inlet and outlet fittings combined in one.

Under service conditions with gear of the telescopic pipe pattern, it is advisable to withdraw the injector pipe every four or five months and examine the packing rings. If new rings are required, the best practice is to turn them out of a piece of ebonite, as this ensures their being a better fit than if standard size spares were carried.

PISTON RODS AND CROSSHEADS.

The piston rod of a single-acting Diesel engine is usually a solid, mild steel forging, attached to the piston by a flange and studs and to the crosshead by a single nut in the usual manner. In some engines the piston rod is hollow, this being to allow of its functioning as a piston cooling medium.

The only gland fitting round the piston rod of a single acting four stroke engine is that which divides the cylinder from the crankcase. It is usual to provide a tray under the piston to prevent any oil or carbon that may drop from it mixing with the lubricating oil. Also, where forced lubrication is used, it is desirable to prevent any oil or oil vapour from coming in contact with the cylinder walls or piston, hence the use of the stuffing box. Such glands are usually packed with metallic rings in sections held in place with springs, and seldom, if ever, need be renewed.

Crossheads are invariably of the square block centre type with a gudgeon pin extending on either side, and the crosshead shoe held in place by studs. The best practice is where the gudgeon pins are thoroughly case hardened and ground to a finish.

With forced lubrication, it is not altogether necessary to have the crosshead shoes lined with whitemetal, provided the bearing pressure is kept low.

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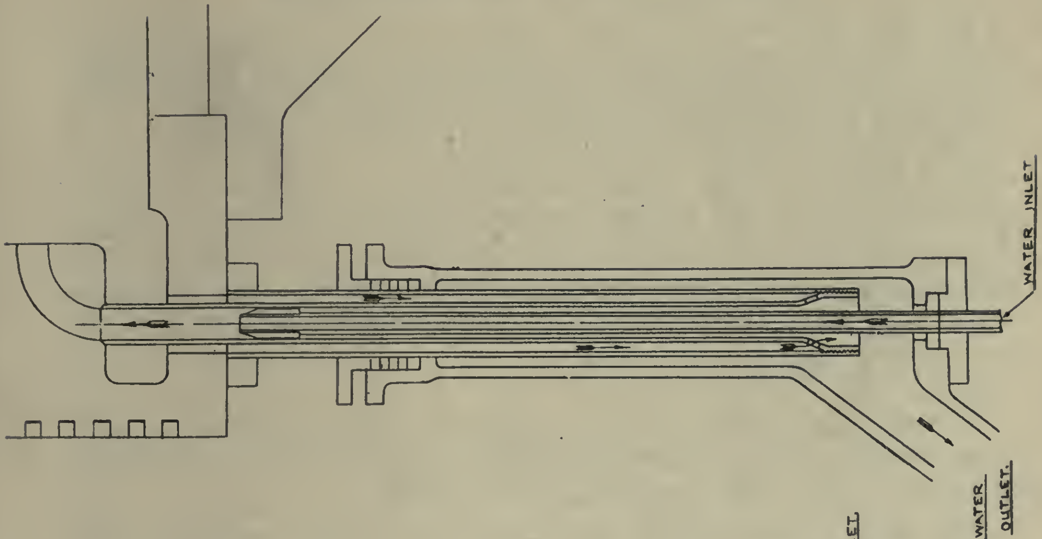

Fig. 6.
Combined Inlet and Outlet Fitting.

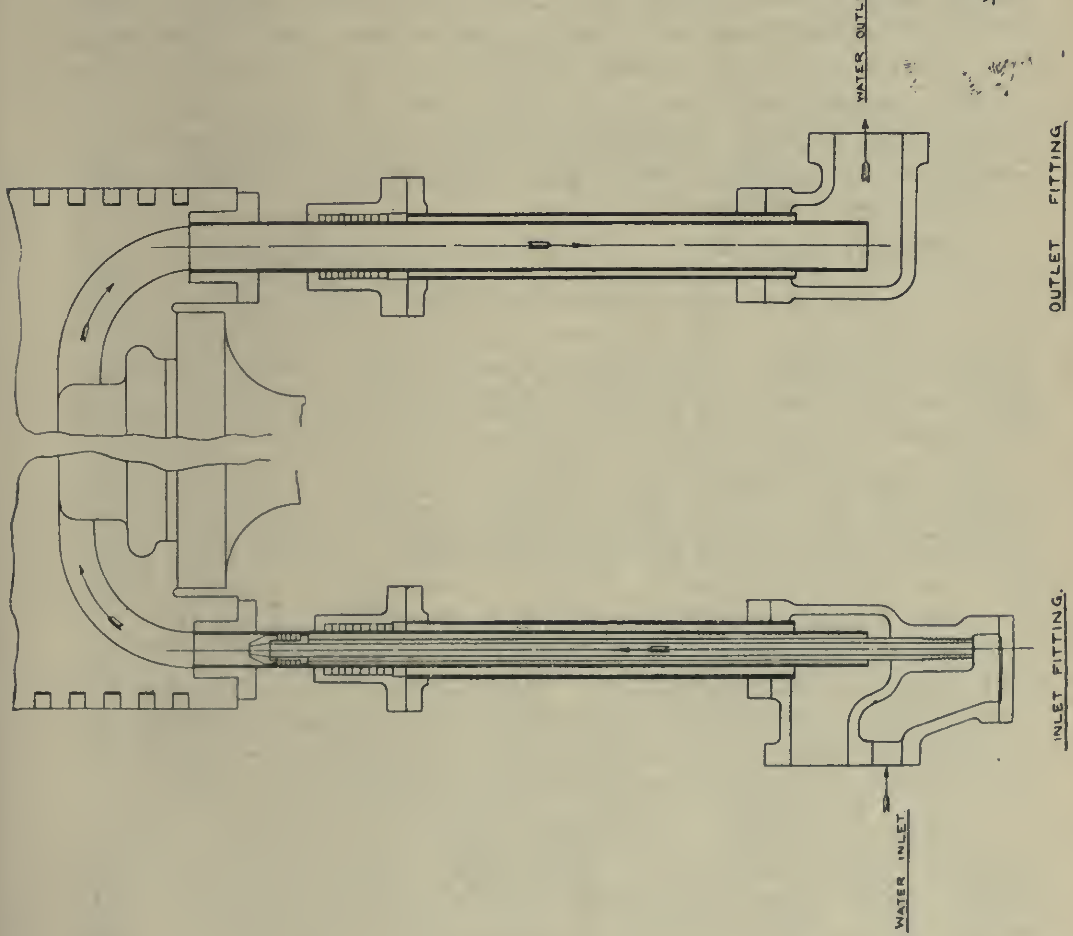

Fig. 5.
Telescopic Pipe Pattern Piston Cooling Gear.

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In the "Beardmore-Tosi" engine (Fig. 1A) the centre line of the cylinder and that of the crankshaft are not co-incident, but are off-set to about 60m/m. This setting is known as "De Saxé" and amongst its advantages are the following:—

The length of the connecting rod can be reduced. The overall height of the engine is also reduced accordingly. Bearing pressures on the ahead and astern guides are equalised, giving low bearing pressures for ahead running. A better impulse is given to the piston at the commencement of combustion.

CONNECTING RODS.

The forked top-end, marine type is that most generally in use. The bushes follow standard practice in that the shells are usually of cast steel lined with whitemetal.

The four stroke cycle forced lubricated Diesel engine shows up much better in service than an open steam engine where top and bottom ends, and indeed, all bearings are concerned. The author's experience has been that where the oil pressure is maintained correct and the filters kept clean, examination of the top ends every six months and of the bottom ends every nine months should suffice.

CYLINDER HEADS.

In the case of a four stroke cycle Diesel engine the cylinder head usually houses five valves, viz., a fuel injection, an air inlet, an exhaust, a starting air and a relief. The principal problem involved is to provide adequate cooling at all parts, so that good circulation of the cooling water must be ensured.

The general practice for securing cylinder heads is to have from eight to twelve studs screwed into the top of the water jacket and extending through holes in the cylinder head with the nuts bearing on facing strips. An alternative to this is to have the top of the cylinder water jacket bulged in the form of a flange of box section suitably stiffened, and allowing of the use of bolts in place of studs. This system obviates the risk of breaking associated with studs. while it gives ease of alignment and also allows the spigot joint between the cylinder head and cylinder liner to be re-ground with a minimum of labour.

In order to simplify construction and to ensure better circulation and cooling effects. it is usual to arrange the fuel injection valve inside a removable liner in the cylinder head. Great congestion results if this is not done. The liners are made of gunmetal or bronze and are either expanded or screwed into place in order to make them watertight. As a method of securing, screwing form the best medium, as then there is no danger of the liner being drawn out along with the fuel valve when the latter is being removed. It sometimes happens that the joint between the fuel valve body and cylinder head leaks, and when this occurs, carbon is formed between the valve body and the liner. The valve is then very difficult to remove, and, when excessive strain is applied to loosen it, the liner, if expanded into place, invariably comes away as well. Similar seizures are caused through water and lubricating oil that may be lying about the cylinder heads, passing down between the valve body and the liner, becoming oxydised and forming a scale which has the same effect as escaping products of combustion. This latter cause can be prevented by placing a rubber ring joint under the valve body head and so sealing the valve port.

As an illustration of the extent of such seizures, the author has seen valve bodies break before loosening, while several hours were spent in removing a single valve under such circumstances. It is imperative that the joint between the valve body and the cylinder head should be made perfect, while it is advisable to draw all fuel valves at least every four months to ensure against any such mishaps.

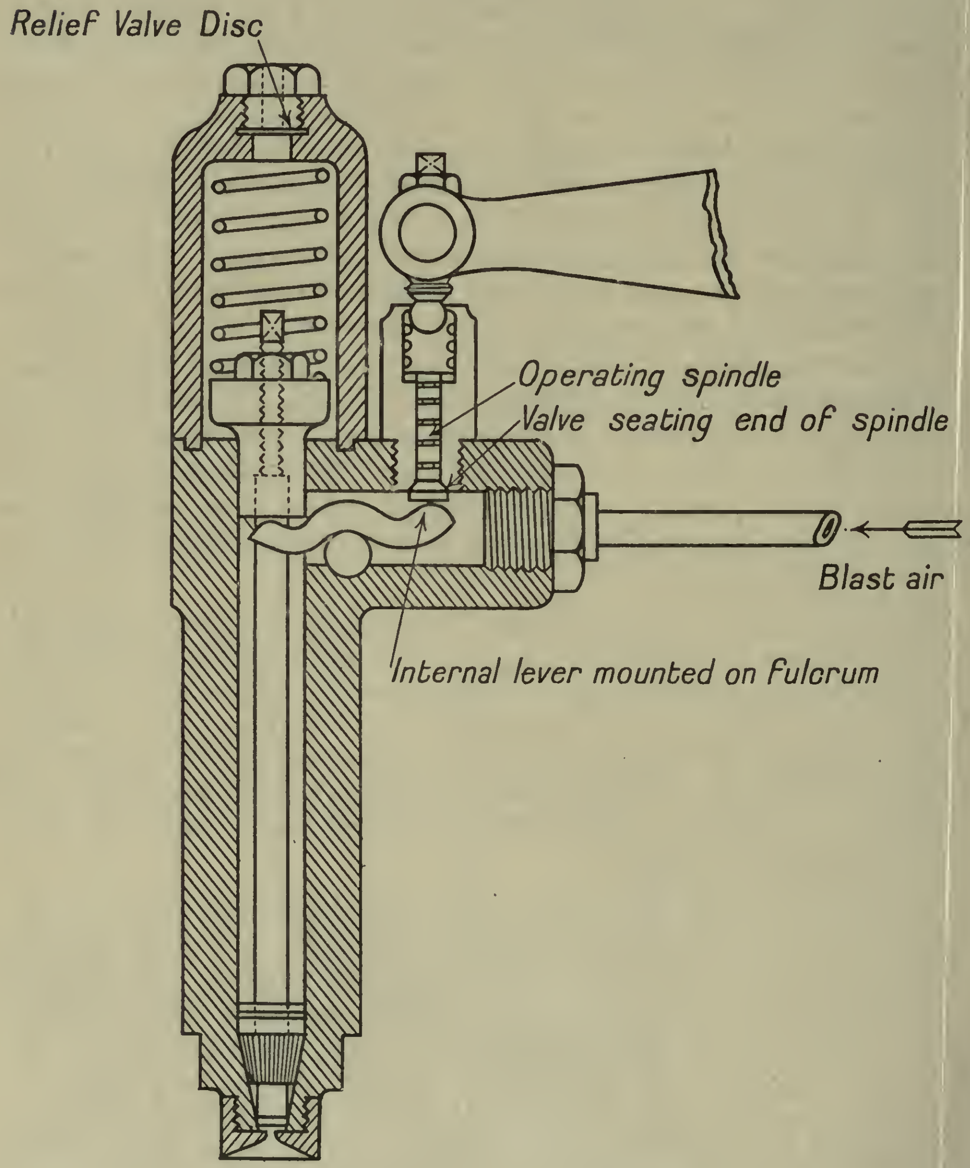
In almost every case the joint between the cylinder head and cylinder liner is formed by a spigot, the male being on the head and the female in the liner. Various modifications of this have been tried, but the most successful is when the joint is made metal to metal with only a little graphite and tallow interposed between the faces. Copper ring joints are too susceptible to the effects of combustion, and as absolute tightness is essential, it is better not to use them.

Cases have come before the author's notice where, due to overheating of the cylinder head joint a slight leak has started, and in developing, has necessitated a stoppage. When the cylinder head was lifted it was found necessary to chip away part of the male spigot and dovetail a new piece in place of it, this also having to be done in the case of the female spigot in the cylinder liner. It is interesting to note that such a repair was carried out with the engine running on its remaining cylinders. It was first stopped and the cylinder head lifted, then restarted and kept running until the head was again ready for lowering into place.

Valves and Valve Gear.

Fuel Injection Valves are divided into two distinct classes, viz., "Mushroom" or downward opening valves, and "Needle"

BEARDMORE-TOSI FUEL INJECTION VALVE.
(PATENT)


Relief Valve Disc
Operating spindle
Valve seating end of spindle
Blast air
Internal lever mounted on Fulcrum

DIAGRAMMATIC SECTION.

Fig. 7.

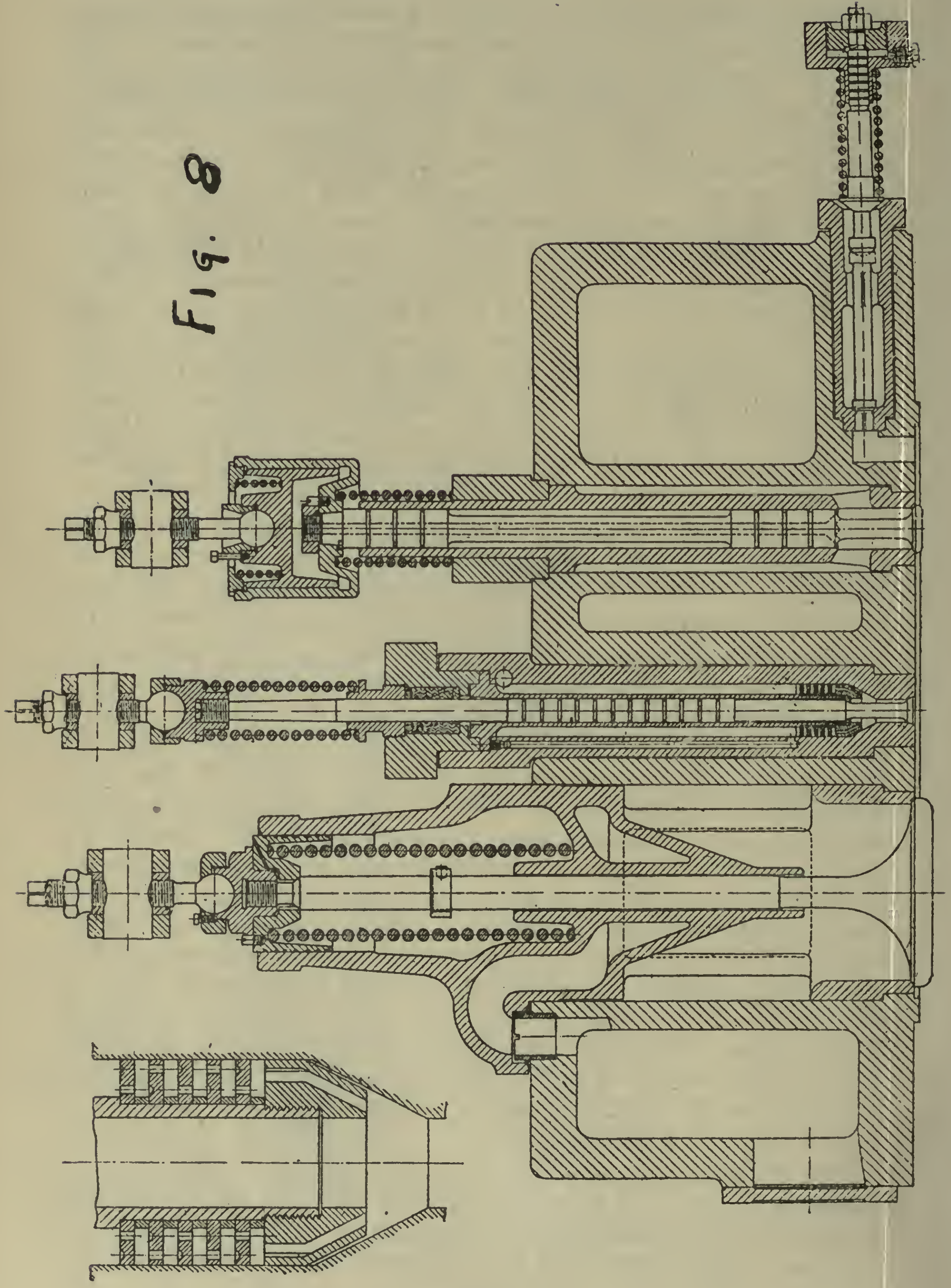
or upward opening valves. There is much to be said in favour of each type, and concrete proof can be brought to bear in support of the claims. The greatest advantage attached to the mushroom valve is that a very evenly distributed spray can be maintained. Further, the spray is in the form of a hollow cone spread over a relatively large area, thus giving every opportunity for good vaporisation. It is possible that this type of valve renders the piston top more immune from the burning action which generally takes place when the oil injected continuously strikes one particular spot. Against these advantages there is the liability of the valve head being burned in the event of the valve sticking open, the effect of which necessitates an immediate stoppage of the engine. Again, it is necessary to withdraw the complete valve when it is desired to examine the seats; also, there is the danger of the valve dropping into the cylinder should the stem break, but this point need not be emphasised as it equally applies to all the valves in the cylinder head.

The needle type has to its advantage (*a*) immunity from burning due to the valve sticking open; (*b*) capacity for being removed and examined, and even re-ground without removing the cage; (*c*) greater delicacy of injection due to increased lift.

The fuel and injection air connections are arranged in the head of the cage and mostly consist of cone couplings. It is general to find the fuel led down through a ¼in. hole in the side of the cage and admitted near the top of the atomiser. The injection air is admitted near the top of the cage well above the fuel inlet. Atomisers are seen in many forms, but the most common is the "plate" type. This consists of a number of plates mounted on a sleeve and kept apart with distance pieces. This sleeve also acts as a guide for the valve stem. Each plate has a number of holes or else slots cut in it, and is so arranged that the holes or slots do not coincide with the adjacent plates. The effect of such an atomiser is to give the fuel a zig-zag motion during the injection period. Fig. 8 shows an atomiser of this type. It will be seen that the sleeve holding the plates has a cone screwed on to the end of it. This cone is provided with special grooves, so that the final passage of oil and air is in the form of a swirl.

The stuffing box shown in Fig. 8 is typical of that used in most fuel valves, and it may be said that packing forms the greatest source of annoyance in connection with this part of the engine. The trouble with fuel injection valve stuffing boxes is

Fig. 8



to keep them tight, and, at the same time, prevent the valves sticking through them. Fuel injection valve spindles invariably show up badly where the packing has worked, even after a short time in service, and nothing much can be done to cure the trouble. The spindles are usually rough and scored and pitting takes place most readily at that section. The author has seen all sorts of packing tried—leather. fibre rings, fibre shavings, whitemetal rings and whitemetal shavings; standard soft packings and standard hard, semi-metallic packings; asbestos cord and plaited lamp wick; a soft pliable mixture of asbestos, rubber, graphite and other ingredients.

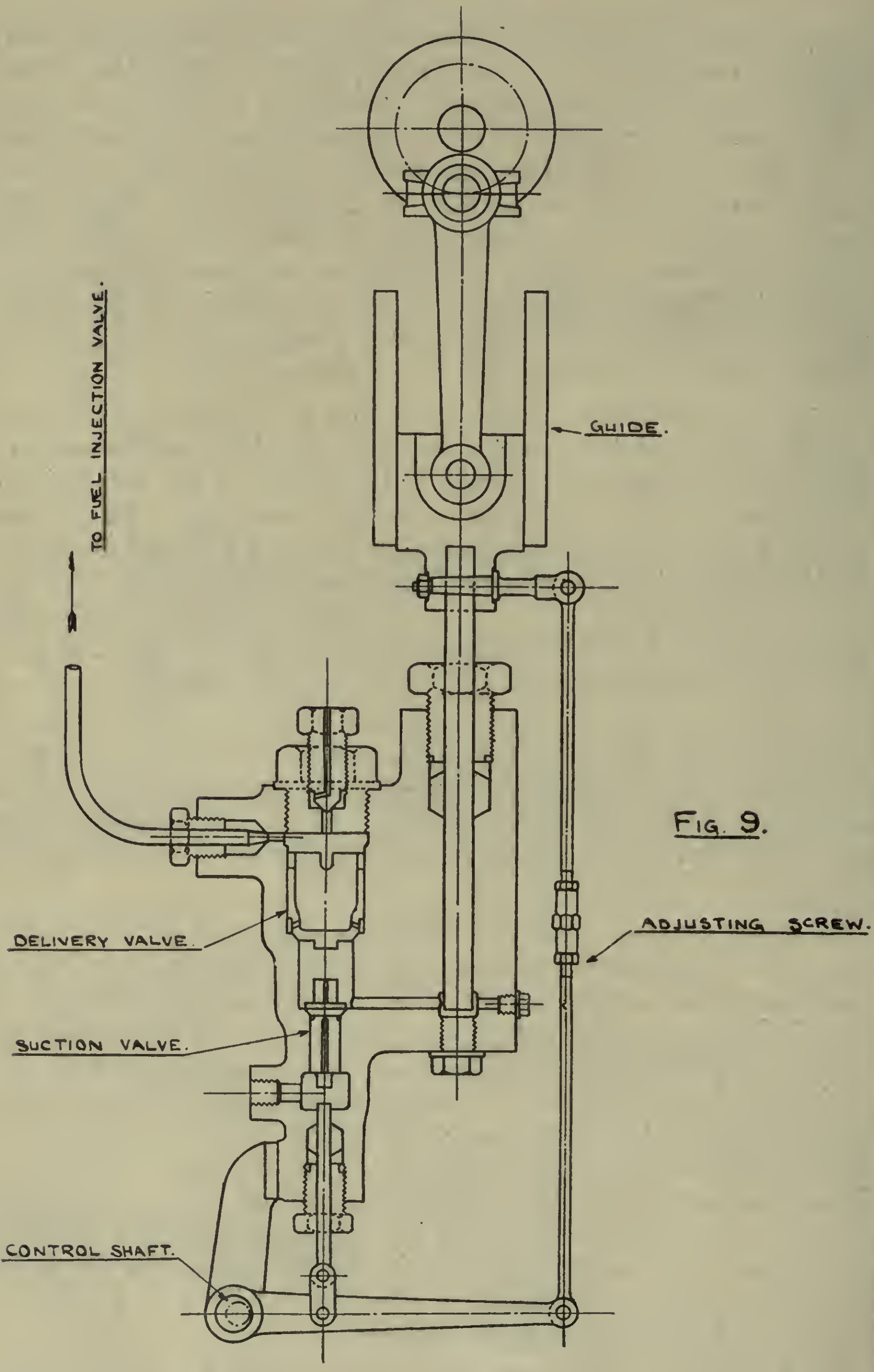
With the stuffing box filled with this last packing and a turn of plaited lamp wick to keep it in place, the best results were obtained, but even these left much to be desired in the condition of the spindles at the end of three months running. The correct solution of the difficulty is to abolish packing glands altogether, as in the case of the Beardmore-Tosi Fuel Valve (Fig. 7).

FUEL INJECTION PUMPS.

As the working balance of a Diesel engine lies in the even distribution of fuel to each injection valve it will be readily appreciated that the main consideration is the distributing gear. To obtain evenness of supply, some makers use a single fuel pump which delivers the oil into a common pipe, and through it to a distribution box, and then to each individual fuel injection valve.

The other system, which is the more all-round satisfactory in service, is to use a battery of fuel pumps each supplying one cylinder. While this necessitates increasing the number of parts, it is amply justified in practice by its evenness of supply and absolute reliability. The adjustment of the quantity of fuel to each injection valve is positive and easily effected. Moreover, in the event of anything happening to a fuel pump, only one cylinder is cut out. Again, if anything happens to a cylinder necessitating the fuel being cut off this can be done without altering the load on the remaining cylinders. Perhaps the greatest consideration of all is the fact that should a fuel pipe burst only one cylinder is affected, while a new pipe can be fitted without stopping the engine.

Fig. 9 is a sketch of a fuel injection pump and, in principle, is typical of most types used in Diesel engine. Differences in design lie in the disposition of the pumps, whether placed



FUEL INJECTION PUMP.

Fig. 9.

vertically or horizontally and in the method of operating them. The most common drive is by means of eccentrics, but in some cases a separate crankshaft is used, it being driven through gearing from some part of the engine. Differences also lie in the number of delivery valves fitted and their accessibility. The best practice is where the valves are fitted in removable cages as this facilitates re-grinding.

The quantity of fuel pumped and therefore the speed of the engine is controlled by varying the timing of the fuel pump suction valves. which are mechanically operated. The suction valves are usually operated by links from the pump crossheads, while the timing is varied by mounting one of the links eccentrically on its fulcrum and by rotating this last member.

Fig. 9 shows the suction valve operation and control, also the delivery valves.

Fuel injection pumps, on the whole, give very little trouble and need little attention. Provided the plunger is well guided and made a good fit in the pump barrel. the packing should last at least six months if the right material is used and the work is done properly. It is usual to examine and re-grind the suction and delivery valves at the same time the pump glands are being repacked. but it is not a very big job and is done easily by a man who understands his business.

Induction and Exhaust Valves.

The valves are all of the mushroom downward opening type, and the only differences encountered are constructional and in the form of seat and cage used. Fig. 8 shows the "Burmeister and Wain" type of exhaust valve from which it will be seen that the valve is set in a removable cage and that the stem guide is watercooled. The valve in this case is turned out of steel and has a removable cast iron seat (not shown). The induction valve used by this firm is similar, in that the same cage is used although the cooling water is not passed through it. The inlet valve is made of steel and has no detachable seat.

The use of separate cages for exhaust valves prevents the seats from being properly cooled owing to their great distance from the water jacket. This necessitates the examination and regrinding of the valves at least every three months, and more frequently if the engine is run at near full power all the time. To overcome this difficulty, some makers use an internally watercooled exhaust valve. for which claims of six months continuous service without examination are made. While the overheating of the seats is to a certain extent overcome by this method, the cylinder heads water spaces still remain unduly restricted owing to the presence of the valve cages.

In the "Beardmore-Tosi" engine (Fig. 1A) a director valve controls the inlet and exhaust. This valve is arranged to work in a port in the cylinder head and controls the flow of gas to and from the combined inlet and exhaust valves by means of ports inter-connecting the inlet and exhaust passages. These passages are alternately brought into communication with the combined inlet and exhaust valve by means of the "director" valve which shuts off either one or the other. Driven by an eccentric, the "director" valve is so timed that as the piston rises on its exhaust stroke with the combined inlet and exhaust valve open communication is made with the exhaust manifold, while the induction manifold is shut off. As the piston passes the top centre to descend on its induction stroke the combined inlet and exhaust valve remains full open, while communication is made with the induction manifold by means of the "director" valve, which at the same time shuts off communication with the exhaust manifold.

The combined inlet and exhaust valve is thus first exposed to the heat of the exhaust gases, but is immediately afterwards cooled by the incoming charge of cold air, further, on the induction stroke, through the action of the air entering the cylinder through the valve, the face is cleared of any soot or deposit that may be lying on it. The "director" valve is situated above and behind the combined inlet and exhaust valve, while accessibility is through a door mounted on the side of the cylinder head. Two "director" valves are fitted in each cylinder head, there being one for each combined inlet and exhaust valve. To ensure the maximum cooling effect, the combined inlet and exhaust valves are made to seat directly on the cylinder head. The advantages gained from this method are such as to render a water-cooled exhaust valve totally unnecessary and to reduce grinding-in to a minimum. Another advantage of the system is that it permits of smaller and lighter valves being used. two being in action instead of one as is the usual practice, thus simplifying the design of the cylinder head.

Fig. 10 shows the operation of the director valve, while Fig. 11 shows the timing diagram.

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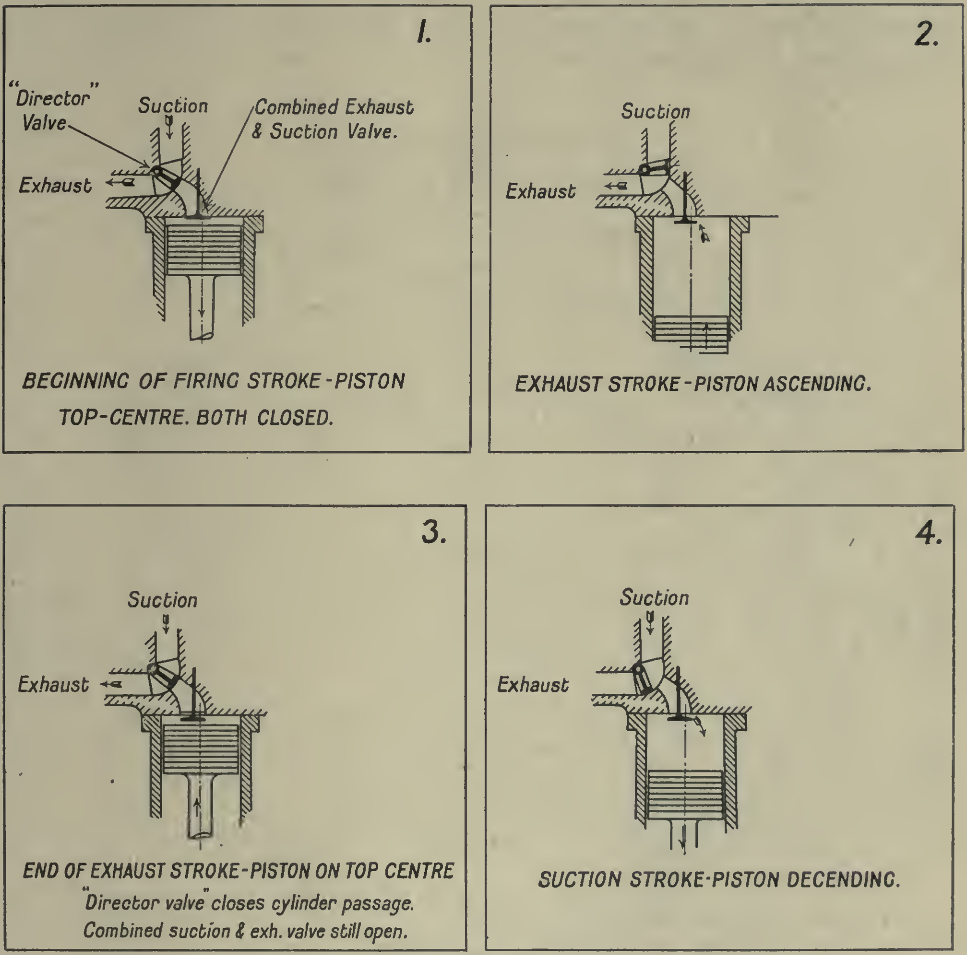
The life of an exhaust valve seat depends greatly on the grade of fuel used, as with bad oil it may be necessary to change the valves every month.

Induction valves can be relied upon for a year's service without re-grinding, and even then they are generally in good condition.



BEGINNING OF FIRING STROKE - PISTON TOP-CENTRE. BOTH CLOSED.

EXHAUST STROKE - PISTON ASCENDING.

END OF EXHAUST STROKE - PISTON ON TOP CENTRE "Director valve" closes cylinder passage. Combined suction & exh. valve still open.

SUCTION STROKE - PISTON DECENDING.

Fig. 10.

STARTING AIR VALVES.

In the majority of engines the mechanically operated type of valve is used, its great claim being positive action. The above-mentioned valve usually consists of a cast iron cage with the valve spindle running through it. Ports are arranged at the bottom of the cage corresponding with the air inlet port at the bottom of the valve pocket in the cylinder head. The valve is made with an ordinary mushroom head, while the spindle is enlarged in diameter above the air inlet ports. From four to eight cast iron ramsbottom rings are fitted, one above the other, on the enlarged diameter of the spindle and serve to make it air-tight. The end of the spindle extends above the cage and is fitted with a retaining cap which holds the valve spring in position against the flange of the cage. The valve is cam-operated through the medium of a rocker lever. Some engines use the

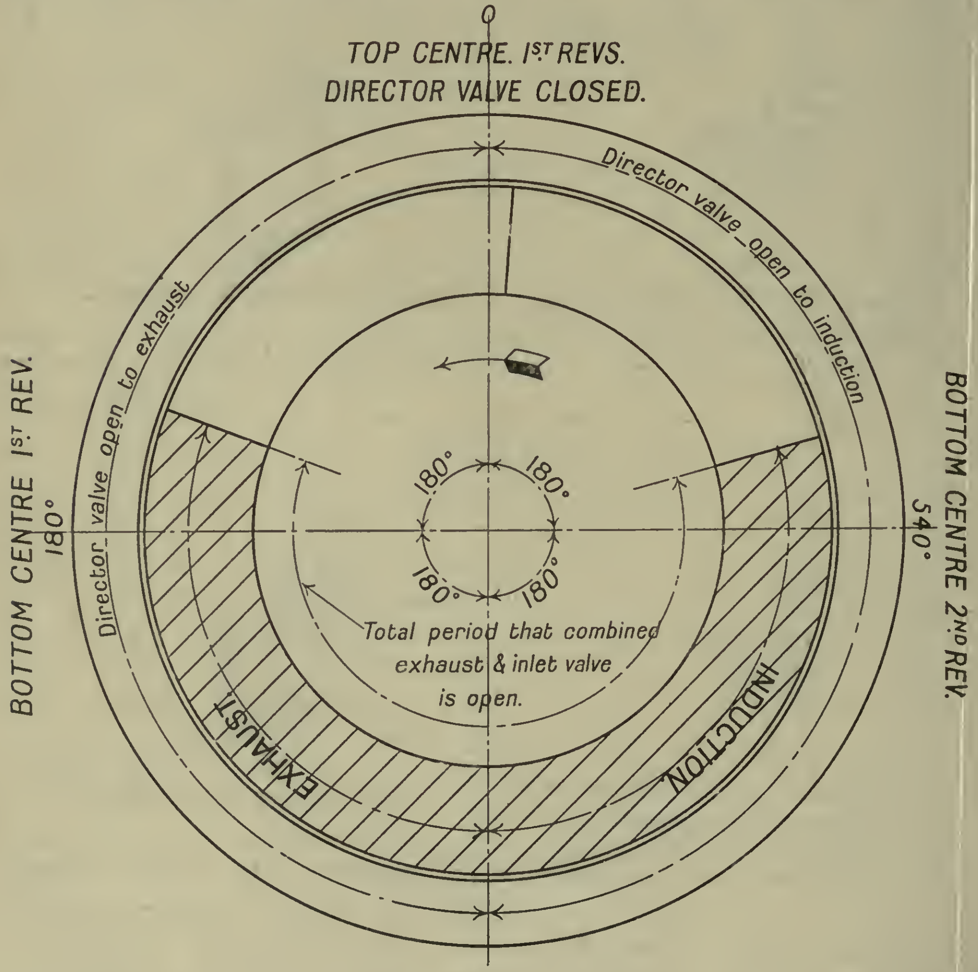


Fig. 11

pneumatically-operated type of starting valve, it is similar to that just described, but is operated by the starting air acting on a piston formed by the enlarged end of the spindle. This type of valve necessitates the use of a timed air distributor driven by the cam shaft or through gearing from the crankshaft. Another system in use at present is that of having non-return valves fixed to the engine cylinders and working in conjunction with a timed distributor as in the case of the pneumatically-operated valve described.

Air starting valves, particularly the mechanically operated type, are perfectly reliable and are capable of long service without needing an overhaul. Their condition can be watched closely while the engine is running on fuel, and, if the valve faces are leaking, indication will be given by the starting air supply pipes becoming hot. Again, if the spring rings on the spindle are also leaking it will be shown by the gas escaping past them becoming visible to the man on watch. Bad leakage from any starting valve would necessitate a stoppage to have it replaced with a spare, but the author has never experienced such a case, it having always been possible to stop the escape by turning and grinding the valve on its seat. To ensure the satisfactory operation of the starting valves, it is advisable to shut off fuel and give the engine a few turns on starting air after every six or seven days running on a long voyage. While this is being done an engineer can watch the behaviour of each valve and if there is any sticking or leakage apparent it can be rectified at the time. Such procedure is also advisible 12 hours before entering any port, as it obviates the possibility of any hitch occurring in the manœuvring of the engine.

VALVE GEAR.

The most usual method of operating the valves of a Diesel engine is by means of cams and rocker levers. In some few cases eccentrics have been adopted, but their use has never become general.

The cams are usually made of special cast iron, ground to a finish and fixed on the camshaft by means of feathers. In existing designs the cam shafts are disposed at the level of the bottom of the cylinders in some engines, and at the level of the top of the cylinders in others. The advantages gained from each position are special, in that the greatest benefits are only apparent when the whole system of valves, valve gear and reversing arrangements is reviewed for each individual engine. Thus, each position has certain disadvantages which are compensated for by other features.

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Different methods are used for operating the camshaft, amongst which are the following:—

(*a*) By a train of gear wheels extending from crankshaft to camshaft.

(*b*) By a vertical intermediate shaft and spiral gear wheels arranged top and bottom.

(*c*) By means of two or more rods operated by eccentrics mounted on a half-time shaft driven from the crankshaft by gearing Fig. 3.

The rocker levers used are invariably of cast steel and differ most generally in the type of tappet adopted. Where a low-level camshaft is used the valves are operated by means of push-rods which have rollers fixed at the bottom, bearing on the cams, while the top ends are fixed to the forked end of the rocker levers by means of swivel eyes and pins. Where a top-level camshaft is used the rollers are mounted on pins fixed in the forked outer end of the rocker levers, thus cutting out intermediate levers and pins.

In four stroke engines two cams are provided for each valve, these being one each for ahead and astern running. The usual arrangement is to place the cams adjacent, but in the early type of "Werkspoor" engine, two camshafts were used, as will be seen from Fig. 3.

Where the top level camshaft is used the usual method of starting and reversing the engine is to mount all rocker levers on a fulcrum shaft, while the fulcrums are arranged eccentrically. By rotating the fulcrum shaft, the roller ends of the levers are raised clear of the cams and the engine is then in position to be set for ahead or astern running. There are several ingenious ways of reversing at present in use, but the commonest is to move the camshaft longitudinally, thus bringing the ahead or astern cams under the rollers in accordance with the desired direction of rotation. When the position is set, the rollers for the induction, exhaust and starting valves are lowered by further rotating the fulcrum shaft, and the engine is put in motion. After making a revolution on air the fulcrum shaft is again rotated in the same direction and three starting valves are cut out, whilst at the same time three fuel valves are cut in. When the engine picks up on fuel the fulcrum shaft once more rotates and the remaining starting valves are cut out, while all fuel valves are brought into action. When stopping the engine the fulcrum shaft is finally rotated until all rollers are again clear of the cams. The complete rotation of the fulcrum shaft is one revolution, while it will be seen that this type of gear corresponds to the "all round" gear on a steam engine. With large Diesel engines it is usual to rotate the fulcrum shaft and move the camshaft by means of air servo-motors, while an emergency hand gear is provided in addition.

The Beardmore-Tosi method of reversing is by mounting the rollers in forks wide enough to permit of their being moved over either the ahead or astern cams. This system obviates the use of a servo-motor to move the camshaft as the operation is carried out easily by hand. The sequence of starting and stopping is the same as that just described.

Where the low level camshaft is used, as in the "Burmeister and Wain" engine (Fig. 2) the system of starting and reversing is as follows:—

The roller ends of the push rods are attached by drag links to cranks on a lay shaft running along the front of the cam shaft. In reversing the engine the lay-shaft is rotated and all push rods and rollers are drawn clear of the cams. At the same time the camshaft is moved longitudinally, this operation being performed by the same servo-motor, and when it is set the rollers are returned to their running position over the cams by a further rotation of the lay shaft. The engine is started by admitting air to the automatic starting valves (Fig. 8) and after a revolution the starting valves are cut-out and fuel is admitted to the fuel valves, the engine then operating normally. It will be observed that the fuel valves are always in action but that no fuel is admitted to them while the engine is running on starting air. This is carried out by making one hand lever control both starting air and fuel, the adjustments being such as to keep the fuel pump suction valves off their seats until the starting valves are cut out.

In most large Diesel engine installations, pyrometers form part of the equipment. These consist of thermo-couples placed as close as possible to the exhaust valves in each cylinder, while the wires are led into an interpole switch and the readings are taken from a common dial usually marked in degrees centigrade. Pyrometers are invaluable for denoting the condition of the exhaust valves, if used in conjunction with indicator cards, while the balance of the engine can be approximated also by their use provided the condition of the valves is known.

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Fuel Injection Air Compressors.

It is usual to find that the air compressors in use are of the three-stage type for large engines. although two-stage and four-stage types are also in service. The principal differences that exist in the various designs are, piston disposition, number of valves per stage and method of operation and control.

The number of valves and variation of type used in each stage differs greatly according to the design. L.P. stages are fitted with from one to five suction and from one to four delivery valves, while I.P. stages are fitted with from one to three suction and delivery valves. The H.P. stages are usually limited to two suction and delivery valves as maximum.

The type of valve used does not matter so long as it is capable of continuous service, but with regard to number the fewer used the better.

With the earliest Diesel engines great trouble was experienced with the air compressors, but this has now almost disappeared. The probable reason is that insufficient attention was paid to the balancing of the stages when running at reduced output, which resulted in increased temperatures being encountered with deleterious effect on valves in particular.

Most troubles now encountered are caused through carbonised lubricating oil adhering to the valves and causing them to stick open. This can usually be remedied without stopping the engine, but it could probably be abolished by the use of proper compressor lubricant and the isolation of the compressor cylinders from the engine crankcase.

Piston rings do not give much trouble if the intercoolers are kept well drained and not too much lubricating oil is used.

Intercoolers vary in design, but the commonest and best practice is where they are of the condenser form, *i.e.*, with tubes and tube plates. The most general trouble that seems to have been experienced with intercoolers is where a coil of piping has been used for the H.P. air. The oxidisation of the metal wears it thin through time. and unless the coolers are kept properly cleaned, bursting may result. For this reason the best practise is where the H.P. cooler is made up of long jacketed straight pipes, as this design allows of ease in cleaning and permits of the bore of the pipes being examined.

Regarding the operating of air compressors, systems differ here as in the other details connected with them. Compressors are driven either by the main engine crankshaft or by beam levers from the crossheads, or else are installed and driven as separate units. Either of the first two systems of operation is preferable to the second. in that the main engine is much more capable of sustaining the drive over long periods than is an auxiliary engine. An auxiliary engine is, necessarily, of lower horse-power and of higher revolution speed than the main engine, and is, therefore, more susceptible to breakdown. Even if the compressors are electrically driven the auxiliary power installed in the ship is greater than is desirable and must be in duplicate to provide against contingencies.

Drawn steel reservoirs are provided between the compressors and the fuel valves and are fitted with shut-off valves. In a carefully arranged system complete immunity from stoppage even through burst pipes is provided for. Blast air pipes are generally made of solid drawn steel with cone-couplings brazed in place.

Starting Air Compressors.

Starting air to the Diesel engine may be regarded in the same way as steam to a steam engine, *i.e.*, each is essential for manœuvring. An auxiliary compressor is always installed, while its power is decided by the special requirements of the ship. Starting-air is usually stored in steel reservoirs at a pressure of from 300 to 500 lbs. per sq. inch. In a single screw ship the auxiliary compressor is usually of the H.P. type, and is made capable of supplying the main engine with sufficient blast air to run it at full power, so that it can be used in the event of the main compressor being incapacitated.

In a twin-screw installation each main compressor is usually made capable of running both main engines at from 80 to 90 per cent. full power, so that no H.P. auxiliary compressor need be installed, as the likelihood of breakdown of both main compressors at once is very remote. The auxiliary compressor installed in twin-screw ships is usually an I.P. machine of the two-stage inverse-tandem type, *i.e.*, with I.P. stage at the bottom and the L.P. stage mounted on top of it, and is just capable of compressing to the starting air pressure.

It is not always necessary to use the auxiliary compressor when manœuvring the main engines, as a great quantity of starting air can be made from the blast-air supply. This is done by opening the main compressor suction intakes to their maximum, thus making the greatest quantity of air, and by slightly closing a shut-off valve that is placed in the pipe-line at the control platform between the blast air bottles and the fuel injection valves. The result is that the blast bottle pressure rises while the blast air pressure drops. A pressure of 800 lbs. per sq. inch is maintained in the blast bottle while the surplus is blown off into the starting air reservoirs. The surplus air that can thus be got usually suffices to manœuvre the engines through the normal operation of docking ship.

Lubrication.

Past experience has proved beyond all doubt that the correct way to lubricate the main working parts of a Diesel engine is to enclose them and supply the oil under pressure. By so doing, a generous quantity of oil is ensured to each bearing, with the result that wear is considerably lessened and the oil consumption reduced. Forced lubrication has been in vogue with four stroke engines for a considerable time now, and in every case has been amply justified. The benefits to be derived are further borne out, in that the most successful two stroke engines have always been forced lubricated. Indeed, drip feed lubrication has been the principal cause of failure of several two stroke engined motor-ships, while in other cases where drip feed lubrication is used and the results, as regards wear, are passable, the oil consumption is abnormally high. The desire to have the moving parts always in view is bought at too high a cost, while the results attained can never justify the price paid. There is very little, if anything, to be gained from having an open engine, because, as a rule, the same indication of any part running amiss exists to practically the same degree as with an open engine. Of course, drip lubricated engines are still being built, but the number has decreased to a minimum and this type is approaching extinction.

The lubricating oil consumption in many motor-ships is high, but there is no doubt that economy could be effected by more attention being given to crankcase and other joints, and to more strict supervision of the quantity of oil used in the hand lubrication of valves and valve gear.

The most general forced lubrication system is to supply the oil from a common pipe through a branch to each main bearing whence it passes into the journal, through the adjacent web and into the crankpin. A central hole through the connecting rod leads it to each top end bush, while the final discharge is to the guides. Such a system can be closely supervised and any choking that my occur can be easily detected.

Before starting on a voyage, the oil is always circulated through the bearings with all crankcase doors open, and, for preference, with the turning gear in. When the oil is seen flowing at all points little anxiety need exist as to subsequent lubrication.

Throughout this paper it has been shown that the marine Diesel engine as it exists to-day, but particularly the four-stroke cycle, is not such an abnormal proposition as it is sometimes made out to be. It certainly consists of more parts than the reciprocating steam engine of corresponding horse-power, but it must be realised that many of the essential features are not subjected to abnormal conditions, and the wear is, in consequence, very slight, being in some cases negligible. When all points of design are considered it will be seen that the engine is not in reality so very complicated. Almost every part can be repaired, so that it is possible to create make-shifts and carry on in the event of a breakdown. Where pistons and the gear connected with each cylinder are concerned, disablement necessitating cutting out the part is rendered easy by virtue of the whole design.

Life on board a motor ship, where the engineers are concerned, is not always a picnic, and in many instances has proved the reverse. There is no doubt but that the work could in most cases be reduced by exercising greater discrimination where examination and overhaul of parts is concerned. With the facilities generally afforded, the condition and behaviour of each part can be ascertained within close limits, so that there is no necessity for working in the dark. The greatest testimony to the Diesel engine is that of the number of motor ships in successful operation to-day. Voyages are undertaken and marine conditions accepted without any trepidation because of the machinery installed.

The CHAIRMAN: There are many items in the paper which struck me, for instance, the long period the machines were running before being opened up for overhaul. It must be gratifying for shipowners and builders to hear such remarks; but there is a word of warning at the end of the paper which causes us to pause, where it is pointed out that the engine will be all right as long as it is properly attended to, and where it is urged that it must not be taken for granted that because we do not see any indication of wrong, all is well within. As the author has said, the internal combustion engines are of 2 or 4-stroke, and there is something to be said for each. It is a matter in which we must be guided by experience gained, or to gain.

A MEMBER: It has been brought to my notice that at times motor vessels have been delayed through engine trouble at different foreign ports. In the majority of cases there has been nothing extraordinary in the trouble itself, it has been due more to the want of repair facilities and experience, also to the absence of spare parts. If the ordinary marine steam engine had been such a new thing or as badly provided for, probably the same delay would have occurred in its case, but now you have excellent facilities all over the world for repairing the steam engine and much experience too.

A MEMBER: The author in his paper refers to a difficulty in obtaining a successful packing for fuel valve spindles; also the numerous types of packings that have been tried to overcome same. I have been away from the type of engine referred to for some time, but with the earlier types, fuel valve and pump packings gave considerable trouble, and once the oil found its way through the gland it was difficult to check, and I think the best policy is to repack the gland as soon as possible. To tighten a gland while the engine is running, care must be exercised to prevent the spindle seizing. Much time could be saved if the design allowed for ample room where possible for repacking and adjustment of the glands.

Regarding the question of piston cooling, the engine I have in mind was originally fitted with a water-cooled system via the crosshead journal similar to the "Stauffer" grease gear. This proved a failure, and was replaced by a lubricating oil cooling system with the oil returning to the crankcase, causing excessive wear in the bearings as the author states, also a high lubricating consumption. Later the oil system was replaced by fresh water cooling with the telescopic tubes as shown in sketch No. 6. I quite agree with the author that the success of the tubes depends upon perfect alignment and calls for close supervision when overhauled, otherwise rapid wear certainly takes place.

A MEMBER: The author speaks of trying the starting air valves before arriving in port, by giving the engines a few turns on starting air. I did not know this was a usual practice. I should like to know from the Author how this ought to be done.

A Member: I have heard a number of engineers with considerable experience express doubt as to whether the Diesel engine did not require more repair than the ordinary steam engine. Can the author give any idea of the ordinary requirements for overhauling a Diesel engine in port, and how it would compare with the steam engine? There is another feature of interest and that is the relative space occupied by a Diesel as compared with a steam engine; also the saving there would be in regard to cargo space. I also believe there is considerable consumption of lubricating oil. It was an item of concern with earlier engines, although I believe economies have been made in later engines. It would be useful to have some idea of any means in design to avoid the use of special lubricating oils throughout, this being a feature which is brought up against the engine—its needing some special grade of oil.

A Member: I, as a steam engineer, find two or three points in this paper that I fail to understand, and no doubt the author will be able to help me. He states that in one design of engine the bedplate sections are provided with male and female spigots which run all round the joints. The presence of the spigots prevents any variation in height of the main bearings, such as is sometimes experienced with bedplates in ships, due to the working of the hull in a seaway. Is it not rather strange to suggest that a cast-iron bedplate, which is not open to bend, can bend to such extent without breaking as to throw out of line the main bearings? We all know in the working of a ship that such a thing as the mal-alignment of a ship's shafting can take place, but up to the present I have not heard of any one engine that suffered from mal-alignment in its crank shaft bearings through the working of the ship in a seaway. The only bedplates I heard of that gave trouble in that way were some of the old marine jobs, that had the condensers in the bedplates and the heat from the condensers, according to the temperature of the water the ship might be sailing in, caused great trouble. I should like that point further explained. Later on it is stated in the paper that in the slow running Diesel engine which is the engine under consideration, sufficiently good balancing is usually obtained by the crank dispositions, and so it is unusual to find counter-balanced crankshafts. If I mistake not, all the crankshafts we saw on the screen were balanced.

The Author: No, not counter-balanced.

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The Member: What revolutions per minute may be termed a slow running Diesel?

The Author: Anything up to 125 or 130.

The Member: Perhaps I misunderstand the diagram. I would like an explanation of what seems to me to be usual experience in steam engine practice, where it is stated as an example of what may be expected from good design and construction, the author's experience has been that with forced lubrication a year's running without adjustment of the guides is quite normal behaviour. There are hundreds of steam engines that are running which have run well without much wear, provided they have not been under-surfaced. In the paragraph under lubrication we read that the desire to have the moving parts always in view is bought at too high a cost, while the results obtained can never justify the price paid. That is a great question we have debated here on more than one occasion, the closed type with forced lubrication. and the open type, with hand lubrication. In regard to this question of forced lubrication, perhaps the author could state what quantity of oil is used per 24 hours under forced lubrication and the cost referred to. I have been chief engineer of a ship of 2,000 I.H.P., and our average consumption of lubricating oil was 4 to 4½ gallons per day of 24 hours for all purposes. Is forced lubrication less expensive than that. The crux of the whole matter seems to come in under the paragraph which says that when all points of design are considered it will be seen that the engine is not in reality so very complicated. Almost every part can be repaired, so that it is possible to create make-shifts and carry on in the event of a breakdown. Is that the style of engine that Marine Engineers want to handle—something that is always liable to require makeshifts in order to carry on? If you are going to sea with an engine that is capable of being continually repaired, I think we have gone from something trustworthy to something exactly the opposite.

A Member: There is a question that I should like to ask the author. When your Diesel engine is running very smoothly and well, sometimes a heavy knock will come on for a few minutes and then disappear again without anything having been done; those present who know the engine seem quite unconcerned about it, but to the uninitiated it is a little startling. The author spoke about the piston top being slightly hollowed or concave in some engines and so collects oil which might go for a stroke or two without burning or exploding, can this be the cause of the occasional knock to which I have referred? Possibly I misunderstood the author.

About breakdowns in motor engines, of course we do hear of these from time to time, and we never hear of the marine steam engine breaking down, or shall I say "Well hardly ever." When I first started going to sea the marine steam engine was not quite the beautiful thing that it is now. It was then no very uncommon thing for it to breakdown. When the motor engine has been on the market as many years as the steam engine has, probably we shall find it as free from giving trouble.

A Member: It seems to me that it would have been much better if the author had given us the benefit of his very valuable experiences early in the session, as it would have settled some of the many points we have only been able to elucidate by someone asking questions and then thrashing the matter out. Anyhow it goes to prove there is no royal road to knowledge, and from the words of our Chairman to-night, with which I quite agree, the Diesel engine has come to stay, and it will be the duty of Marine Engineers to educate themselves up in the working of the internal combustion engine, the application of this class of engine bristles with difficulties, and with the great saving effected it has many points to recommend it to shipowners. But what the price of oil will eventually go up to, it is very difficult to say. Therefore with ships burning oil or coal, which could always be obtained, ships would be safe to go to any part of the world.

This has been very amply proved by the British Navy having the ships mostly fitted by oil-fired boilers which could be quickly altered to burn coal as required. The test for the internal combustion engine is, has it reliability on service? The author has stated that the main engines gave very little trouble while the auxiliaries always keep the engineers busy off watch, and he has promised to tell us more about these in a future paper.

The Chairman: You must remember, those of you who were present at the general meeting when our new President came before us for the first time, that he touched on this subject in rather a specific manner. He drew attention, as a business man, to the phase which the country was passing through, and might be passing through for some little time. You are all acquainted with the circumstances prevailing to-day unfortunately, and if ship-owning is to be run as a paying business, as all businesses should be, on their own basis and on their own bottoms without State aid, our President pointed out clearly that he and other shipowners would have to look round and see whether they were getting the tools that would give the best economy. Competition is the soul of business, and it behoves every man when he starts in business to see that he has the right plant and the right men to bring about that success. Our President spoke of the internal combustion engine, and said that it had been recommended to him. He did not speak as an engineer, but as a shipowner, whose interest was in his plant. He referred to the resources here for the proper economical working of ships. We know that whilst there are ships there must be engineers, and he drew particular attention to the possibilities of the future of the internal combustion engine. Of course, there is a long way to go yet before we see the steam engine scrapped. There is a sense of reliability and confidence begotten of long years and practice in the use of the steam engine, and when any new venture comes along it is quite a time before it becomes universal. One cannot jump from the one to the other. because whatever has been the experience, we have not the men prepared to take it up with that confidence, that would justify it in fairness to themselves. It takes time to educate men, and it takes time to get all full information in regard to types of prime movers. We can see from the remarks of our President what the commercial world is thinking, and to what end they are turning their attention. He spoke of oil fuel, and in quite another phase to the internal combustion engine. Whether the oil fuel steamer, with the necessary personnel and the number of men you have to employ would be most economical, or the internal combustion engine, where some costs are eliminated, and it becomes a question of engineering ability. Given that the steam engine is running all right, the main thing is to get steam to push the ship along. In the Diesel engine that is eliminated, and you have to give all the skill and knowledge to the care of the engine. As to the bedplate, I do not think the author exactly meant what has been implied, but he spoke of some features of the bedplate of this engine as to which I was a little surprised. Sometimes you find bedplates. if they have not been properly wedged with iron as well as wood jacks, that they will go out of line and give you severe trouble. But if it comes to a trouble in a seaway there is something wrong. In the latter part of the paper I think he meant to show that the Diesel engine was not such a terrible affair after all, but that it was possible to repair it; not that it was going to stop every now and then, so that the engineer would have to keep at it daily. He wants to inspire confidence in those who perhaps look at it from another point of view from that point of view which he had in his mind. A member spoke in regard to the relative space occupied, and that drew me back to the remarks of the President. Anything that is to be put forward to-day in place of the present steam engine, whether it be turbine, or gearing, or electric drive, it all comes back to one thing—what can a shipowner obtain in the way of space? Can he carry more cargo and so afford to pay more money for extra plant? How much space will this internal combustion engine occupy that will justify him in making the changes, and is it worth it from an economic point of view.

A Member: Do I understand rightly that the air compressing gear on these Diesel-engined ships is supplying sufficient air for working the engine, say for 30 minutes, or even off and on while docking ship and the necessary movements of the engines, do I understand these movements are carried out by compressed air only and that there is no fuel supplied to the cylinders to work the engine, but it is all done by compressed air? Reference was made to the packing, and I see lamp wick is one of the things mentioned. Has the author tried cotton packing, the cotton packing we have at the present day for the ammonia machine? Then I am also interested about taking the inner details out of the engine; that is, getting the piston out and in. There seems to be a lot of intricate dismantling, and then it is looked upon as an easy job, as you have no boiler work. Is there any other method of designing a piston so that it can be drawn from the top, or, is it always in all cases like this, drawn from crank end of cylinder? Apparently in some engines the piston is taken from the underside of the cylinder. And I should also like to ask the author if he is a two-stroke man or a four-stroke advocate.

A Member: Can the author give the relative cost per horsepower of the Diesel-engined ship as compared with the steamer? Are there any comparisons of upkeep and maintenance costs? I think these are very important points, and we should like to have figures, if available.

The Author: In dealing with the rapid advance of engineering to-day, the Chairman emphasised the necessity for the Marine

Engineer making himself familiar with current progress. Since the successful operation of the machinery on board ship depends on the engineers, and, as the number of motor ships in service is increasing rapidly and the engine rooms have to be manned, then familiarity with the Diesel engine becomes imperative to all ambitious Marine Engineers.

The Chairman remarked on the long interval between examinations of parts of the Diesel engine, but also noted the necessity of strict attention being given to the running. As I state at the end of the paper, facilities are generally provided whereby the condition of the engines can be closely ascertained, and by the proper use of such facilities the work of examination can be reduced.

A member mentioned having heard of motor ships having been delayed in foreign ports through engine trouble, and that the trouble was, in the main, due to inexperience, also to lack of facilities and the absence of spare parts. In my reply to one member, I emphasised the necessity for Marine Engineers being familiar with the plant under their care, because, if they are not, then troubles are bound to accrue. Regarding facilities for repairing Diesel engines, and the amount of spare gear carried on board a motor ship, my experience has always been that the amount of spare gear carried is usually sufficient to overcome any troubles likely to be encountered on any one voyage, while there are very few parts of a Diesel engine that cannot be copied and remade by any ordinary engine repair shop. It is to this question that I refer in the second last paragraph of my paper. In proof of my statement regarding replacements, I have known an engineering firm in China replace the cylinder head of a four-stroke auxiliary Diesel Engine of which they had to make the pattern, cast it and machine it, and finally get it passed by Lloyds, while the whole of the work was done by Chinamen.

The experience referred to with a motor ship goes further to prove that the Diesel engine is not an abnormal proposition, because the troubles encountered were apparent, and, I take it, were easily overcome once the work was understood.

A member refers to troubles experienced with the packing of fuel pumps and fuel injection valves. Well, of course, the proper remedy is, as stated in my paper, to abolish packing glands, where the fuel injection valves are concerned at any rate, and I can assure him that this can be done successfully. Fuel pump glands are in no way comparable with the glands of fuel injection valves, as the packing gets such different treatment in each case. The fuel pump plunger is usually of relatively large diameter and has a good long stroke, while the packing is exposed to fuel pressure only. The fuel injection valve spindle, on the contrary, is relatively small in diameter, while its movement is limited to a 5/16ths inch lift as maximum. Then again, the gland has to remain tight against air, as well as oil pressure, and when it is considered that the opening of the valve approximates to a hammer blow, it will be admitted that packing of any sort is severely tried. This member's experience with piston cooling gear bears out my statements, even to the dissatisfaction of oil cooling.

A member asks how starting air valves should be tested before entering port to ensure satisfactory operation. With the "all-round" gear described in the paper, all that is necessary is to open the starting air stop valve and set the servo motor in action, when the valve gear will go through its complete sequence, which includes all starting air valves in operation. With the "Burmeister & Wain" Engine (Fig. 2) the test is carried out by pulling the control lever to "stop," then putting it to the "start" position, which puts all starting air valves in action, and, when the engine has made a few revolutions on air, returning the lever to the "on fuel" position. In carrying out this test with any valve gear, the engine never stops, as the way on the ship carries the propeller round.

Regarding the question of maintenance and overhauling requirements in port. Throughout the paper I have given the service time that may be expected from each part of the engine, and the overhauling of each part is not frequent. The examination and overhauling time for each part being known, the Chief Engineer arranges matters so that the work is divided, in proportion to the duration of stay, over the ports of call. In this way it is possible to execute all main engine adjustments without the need for working overtime, just as in a steamship. Any troubles that are encountered with a motor ship are usually more due to the auxiliary rather than to the main engines.

Regarding the relative space occupied by Diesel as compared with steam machinery, this question depends entirely on the installation arrangements. With good design and layout a motor ship can have 5% more cargo space than a similar sized steamship, in addition to a much greater cruising radius.

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On the lubricating oil question and in particular as to running costs, I would refer you to a paper read by Mr. James Richardson, before the Institution of Engineers and Shipbuilders in Scotland, and entitled "The Present position of the Marine Diesel Engine." In it he quotes as follows:—

TABLE V.

COMPARISON OF RUNNING COSTS OF DIESEL SHIPS AND STEAM SHIPS OF 2,400 BRAKE HORSE-POWER.

	Twin-screw. Diesel, 2400 B.H.P.	Single-Screw, Double-Reduction Geared, 2400 S.H.P.		Single-Screw, Reciprocating, 2800 I.H.P.	
		Coal.	Oil.	Coal.	Oil.
Fuel, lbs. per H.P. per hour	·45	1·5	1·1	1·75	1·25
Consumption, tons per day	11·6	38·6	28·3	52·5	37·5
Consumption, tons per 30 days	348	1,158	849	1,575	1,125
Price of fuel per ton ...	£11	£5	£10	£5	£10
Cost of fuel per 30 days	£3,828	£5,790	£8,490	£7,875	£11,250
Lubricating oil consumption, gallons, per day	21	3	3	5	5
Lubricating oil, cost per gallon	5/-	5/-	5/-	5/-	5/-
Lubricating oil, cost per 30 days	£157 10	£22 10	£22 10	£37 10	£37 10
PERSONNEL—					
Chief Engineer ...	1	1	1	1	1
Assistant Engineers	6	3	3	3	3
Greasers	3	3	3	3	3
Firemen	—	9	6	9	6
Trimmers	—	4	—	4	—
Donkeyman ...	1	1	1	1	1
Electrician	1	—	—	—	—
Total engine-room staff	12	21	14	21	14
Total wages, 30 days...	£252 10	£380 10	£265	£380 10	£265
Total keep per 30 days, at 7/- per day	£126	£220 10	£147	£220 10	£147
Total wages, fuel oil, and keep for 30 days...	£4,364	£6,413 10	£8,924 10	£8,513 10	£11,699 10
Ratio	1	1·47	2·04	1·95	2·68
Net saving per annum of 200 days sailing, Diesel over Steam ...	—	£13,650	£30,390	£27,645	£48,850

NOTE.—In addition to the above, the following savings are effected. Fuelling costs, less demurrage, additional cargo capacity, less accommodation for engine-room staff, no stand-by losses, less cleaning ship, higher average-speed in a ase-way, reduced fuelling appliances required, etc.

TABLE VI.

COST OF FUEL OIL AND COAL AT PRINCIPAL PORTS, JULY, 1920.

PORT.	FUEL OIL PER TON.	COAL PER TON.
Alexandria	250/-	186/- to 200/-
Adelaide	180/-	40/-
Batavia	150/-	127/-
Bombay	150/-	45/-
Buenos Aires	265/-	170/-
California	62/- to 92/-	69/-
Christiana	224/-	210/-
Calcutta	250/-	25/-
Cape Town	220/-	46/9 Transvaal.
Colombo	150/-	102/6
Curacao	80/-	125/-
Glasgow	250/-	115/- Welsh.
Hong Kong	150/-	115/- Welsh.
Havana	142/6	125/-
Karachi	150/-	45/-
London	250/-	115/- Welsh.
Liverpool	250/-	115/- Welsh.
Lisbon	250/-	160/-
Madras	150/-	45/-
Melbourne	180/-	35/-
New Orleans	58/2	40/- to 69/-
New York	47/6	55/- to 68/-
Palembang	125/-	127/-
Pensacola	80/-	42/-
Port Said	250/-	186/6
Panama	75/-	125/-
Rotterdam	220/-	160/-
Rio de Janeiro ...	250/-	185/-
St. Thomas	160/-	124/-
Sydney	175/-	21/3

Average price per ton :—Coal = 104/- ; Fuel Oil = 191/-

The heavier lubricating oil consumption of the Diesel engine is accounted for by the quantity used in the cylinders and air compressors. The bearing oil consumption is much less than that of an open type drip-feed lubricated steam engine in which the bearing oil used is not recovered.

The brand of lubricating oil used in the Diesel engine does not form a problem in need of a solution. For a forced lubricated engine, any straight mineral oil will give perfect satisfaction, and one grade can serve all purposes. Turbine oil, which is a straight mineral product, is procurable at all important centres in the world, so that the motor ship is in no danger of running out of supplies.

A member takes exception to my statement regarding the distortion that can take place in a Diesel engine bedplate, and quotes his experience of steam engine bedplates in comparison. In the first place, the bedplate of a Diesel engine is considerably longer than that of a steam engine of corresponding horsepower, and, of course, as the length increases, so does the tendency to distort when the hull of the ship is working in a seaway. Apropos of this, it will be found that in the latest "Werkspoor" engine, wedges are provided under each main bearing and provision is made to adjust tnem, by having a long bolt attached to each, passing out on the front of the bedplate.

Regarding the duration of service for guide plates, his view is that a year's running without adjustment is less than ordinary behaviour for even a steam engine. Well, I have often had cases of steam engines brought before my notice, where attention to the guides was required at least every six months, and the wear during that period was considerable. Of course, the trouble may have been due to lack of bearing surface, which, as he grants, can cause excessive wear.

He states that the question of the totally enclosed, force-lubricated engine, as opposed to the open type with hand lubrication has been discussed here on many occasions, and that it still remains a debatable point. As I have stated in my paper, past experience gained with Diesel engines would appear to leave very little doubt as to which is the better system of lubrication, and, if examination is made of the various designs of engine building to-day, it will be found that the vast majority are force-lubricated. In his remarks on the lubrication economy of the steam engines, he asks if force lubrication could show an improvement on the figures he quotes. Well, if a comparison be made between hand and force lubricated *steam* engines, then I think it will be found that the latter type will show a saving of 80%.

In his concluding remarks, this member has, I am afraid, sadly misinterpreted my statement regarding the degree of reliability of the Diesel engine. I fail to see how any statement in my paper can convey the idea that the Diesel engine is in need of continual repairing, far less being "something that is always liable to require makeshifts in order to carry on."

THE MARINE DIESEL ENGINE: ITS RELIABILITY IN SERVICE.

When I wrote this paper and took each engine part in detail, I wanted merely to give a survey of all the parts involved and to show that the design of the Diesel engine is such, that should any part break down it is possible to repair it. The reliability of anything is ensured from the point of view of design, and while the engine should be capable of long and continuous service, we cannot guarantee that there is not the remotest chance of its ever breaking down. Then, so long as that doubt remains—as it must always remain—absolute reliability is only ensured if it is possible to make repairs.

The knock sometimes occurring, referred to by another member, would probably be caused by a fuel injection valve sticking open momentarily, the fault lying in the packing being too tight. Sticking is an inherent trouble with nearly all fuel valves which use packing, and its effect is to raise the compression in the cylinder to an abnormal degree, which results in an augmented explosion pressure and consequently a heavy knock is heard. The compression is raised by the blast air continuously entering the cylinder, and while its effect is negligible during the exhaust and firing strokes, it raises the initial pressure at the beginning of the compression stroke with the results I have already mentioned. In connection with such an occurrence, a further claim can be made for the system of providing a separate fuel injection pump per cylinder. When this is done it is impossible for any one cylinder to get more than its measured quantity of fuel, so that, when the fuel valve sticks open, the other cylinders are not starved as would be the case if only one or two pumps were fitted and all the fuel valves were connected by a common pipe. Further, where the "common rail" system is adopted and a fuel injection valve sticks open, the resultant explosion pressure will rise tremendously, due to the unlimited quantity of air and fuel entering the cylinder. The effect of misfiring, which I state in the beginning of my paper, is similar, but the pressures involved are not so high.

The remarks regarding the steam engine are intensely interesting. Improvements have taken place so rapidly in design, and already so much experience has been gained, that I think it may be truthfully said of the modern 4-stroke Diesel engine that one hears of break-downs. "Well. hardly ever!"

I can hardly agree that the application of the Diesel engine "bristles with difficulties," because, if that were so, it would scarcely recommend itself to ship owners as he grants it does.

A member mentioned the use of cotton packing with the ammonia engine. As I have said, it is not so much a question of packing as the service it has to perform. Then in respect of the removal of pistons—at the beginning of my paper, I enumerated the various designs of cylinders. In the majority of cases, to remove the piston, the cylinder head is lifted and the piston is drawn up after removing the piston rod nut. In the "Werkspoor" engine the pistons are removed from the bottom of the cylinder, while in the "Beardmore-Tosi" engine either system is applicable.

The question was raised as to whether I am an advocate of the two or the four-stroke engine. As matters are at present, I am an advocate of the four-stroke cycle.

The Chairman spoke of the possibilities of the Diesel engine. Well, of course, progress with the Diesel engine, like everything else, must be slow and sure. Our safest action lies in slow and careful development where each size of cylinder is tried out, and the many problems involved are carefully solved. As we advance slowly we advance surely. Where big engines have been tried, engines of 2,000 B.H.P. per cylinder, I am afraid they were started at a date when there was insufficient knowledge of the Diesel cycle to warrant such an experiment, with the result that the data derived has not been of any particular service to Diesel engine designers.

The Chairman raised the question as to the training of engineers and the time it would take to get steam engineers acquainted with the motor engine. With the advent of the Diesel engine, a demand arose for the supply of engineers capable of handling the machinery, but the difficulty was met by steam engineers volunteering for service. After having from six months to a year's experience in the shops in the erecting and testing of the engines, they were capable of going to sea under a guarantee engineer supplied by the firm, and so the engine rooms were manned and the ships operated very successfully. To a man of average intelligence, and one who can adapt himself, there is no great problem involved in the training, whilst, of course, presence of mind and readiness will stand for any machinery and anything that is encountered.

The Chairman spoke of having steam engines running all right, and the trouble of keeping a steady head of steam.

I recall once going through the Red Sea in an abnormal temperature. We were rated as an eleven knot ship, and we passed a liner rated at thirteen knots, while it took her several days to catch us up in the Indian Ocean, the reason being, I take it, that it was practically impossible for the steamer to keep up a steady head of steam, owing to excessive heat in the stokehold. With Diesel engines we maintained steady revolutions without needing any adjustments whatever. Even when running through heavy weather very little variation is noticed with Diesel engines. The revolution speed has a tendency to drop and the M.E.P. to rise as with the steam engine, but all that is necessary is to shut in the fuel control lever should the M.E.P. become too high. A safe figure to maintain with the four-stroke engine is 90 lbs. per sq. in. I H.P. basis. It will be found that the Diesel engine maintains a steady average speed over a voyage including varying weather conditions.

On the question of the relative cost of steam and Diesel machinery. Quoting Mr. Richardson's paper already referred to, he states that at present the cost for Diesel machinery varies from 25% to 30% more than for a steam plant, depending on the type of auxiliaries applied to the oil-engined ship, and whether comparison is made with reciprocating or double-reduction turbine steam machinery. He also shows the great saving effected by the use of Diesel machinery, of which the table I read is an example.

A Member: A question has been raised about the advisability of trying the air for manœuvring before going into port; that is very like what we do now with the steam engine. We should not think of approaching port without seeing that the steam reversing gear had been heated up and moved backwards and forwards to free it and make it handy. As to the question of reliability, the fact that a motor ship owner will accept any charter that is acceptable to the owner of a steamer and also that the charterer will accept a motor vessel, readily shows their point of view on the matter.

The Author: I should have liked to treat the auxiliary Diesel engine on board the motor ship more fully, but of course it was not possible within the scope of this paper. It is with that engine that perhaps most of the troubles have been encountered, even right from the beginning, while the tendency is that it always will be so. It is just a parallel case with the steam ship where it is not the main engines but the auxiliaries that keep everyone busy and create field days.

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Mean pressure on indicated horse power basis, correctly termed mean indicated pressure, is the figure used in calculating the I.H.P. of the engine. Mean pressure on brake horse power basis, correctly termed mean effective pressure, is a fraction of the mean indicated pressure, and is dependent upon the mechanical efficiency of the engine.

e.g. M.I.P. = 100 lbs. per sq. in.
Mechanical Efficiency = 75 °/ₒ.
∴ M.E.P. = 75 lbs. per sq. in.

In further reference to the point as to temporary disablement of an engine; suppose, in a Diesel engine, that a crack should develop in one of the pistons, then all that would be necessary to enable the engine to carry on would consist of cutting out the fuel pump for that cylinder and shutting off the piston-cooling water by means of the regulating valve provided for the purpose, all of which could be done without stopping the engine. Suppose a valve in the cylinder head should break or seize, then cutting out would be equally simple, although, in this case, it might be necessary to first stop the engine. The very worst accident happening to a cylinder or any of its gear, could only entail removing the connecting rod and then the engine could be re-started.

This is not so with a steam engine, where compounding as a makeshift in the event of a cracked cylinder cover or like accident, entails considerably more work and a longer stoppage.

SESSION 1920-21.

President: LORD WEIR OF EASTWOOD.

The Solid Injection Engine.

BY MR. C. MCTAMINEY (Member).

Read Tuesday, October 26, 1920.

I was very interested to read the paper on solid injection Diesel engines published in the proceedings of February last, and as the subject seemed to be of great interest to the members generally, I thought that perhaps some information bearing on the use of solid injection in larger engines might be appreciated, as the previous paper dealt more particularly with the submarine type of engine. Having been from May, 1917, till quite lately, in charge of the machinery of the *Trefoil*, an oil tanker running as a Royal Fleet Auxiliary under the Admiralty, I gained considerable first-hand experience of the running of these larger engines.

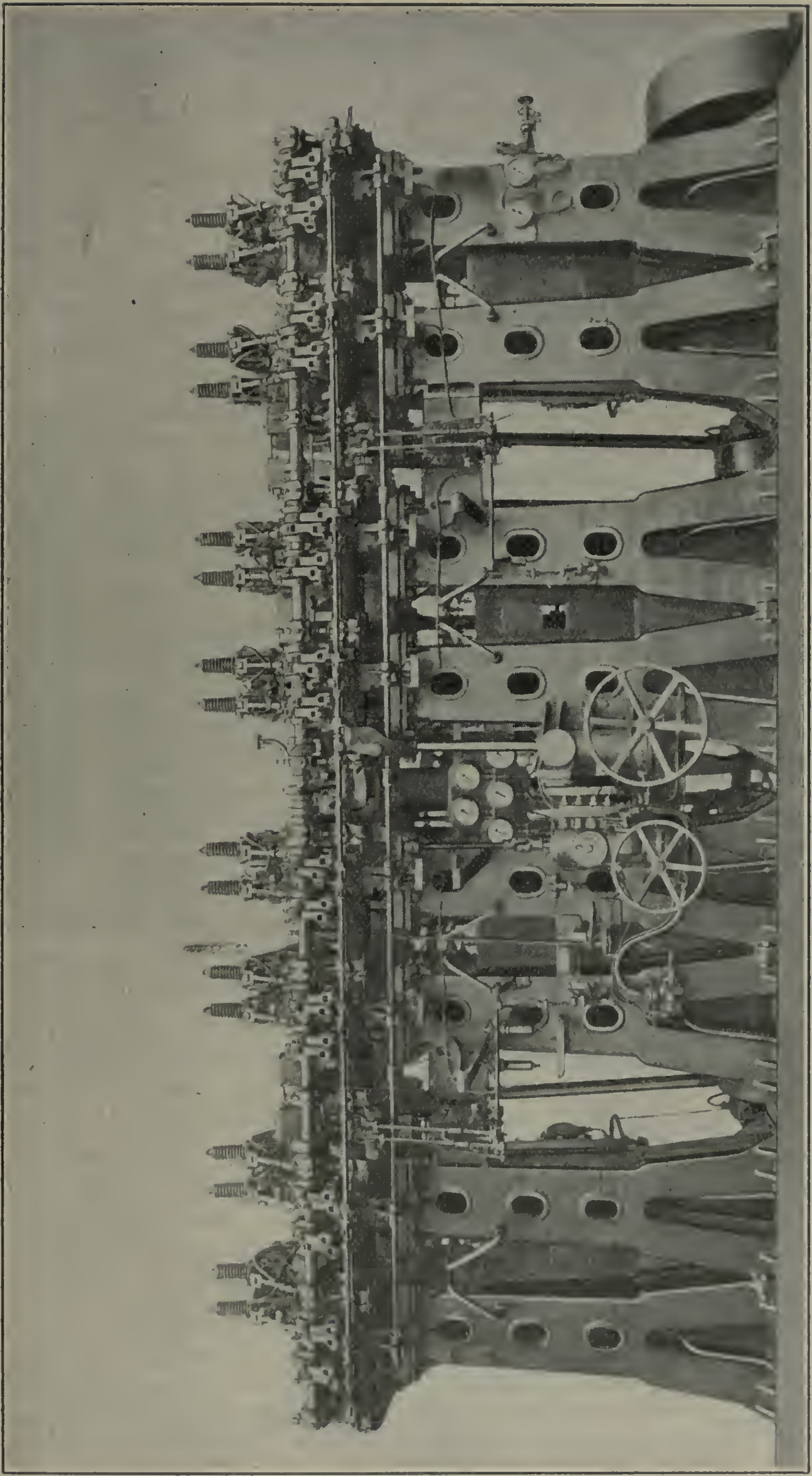
The main engines of the *Trefoil* are of the four-stroke cross-head type, the vessel being fitted with two Vickers' engines, each of eight cylinders and together developing 1,500 brake horse-power at 150 revolutions per minute. The vessel herself carries 2,000 tons of fuel as cargo on a displacement of about 4,510 tons, her speed being 11·5 knots at full power and capacity of bunkers 213 tons.

These engines, it may be interesting to note, are the second pair made for the *Trefoil*, the first pair built for the ship having been diverted to the Monitor *Marshal Soult*, one of the ships hastily constructed for work on the Belgian coast. The *Trefoil* was fitted with a replace pair of engines, by the same makers, generally similar to those in the *Marshal Soult*, but with minor improvements.

The original engines formed one of several Diesel marine installations which the Admiralty ordered before the War so as to obtain a comparison of the sea-going merits of the different makes which were on the market. They are not pure commercial engines, as with an eye to the developments for warships, it was arranged by the Admiralty that they should be of the low headroom type, and the construction was based on the use of a stepped bedplate—as is common in warship work—together —in the *Monitor* at any rate—with forged columns braced somewhat as in torpedo boat construction, but on a larger scale. As a matter of fact, the second pair of engines had cast columns on the inboard side in order to cheapen cost and facilitate construction, it not being necessary to repeat the experiment in the second pair of engines.

The general construction of the engine is plainly shown on the drawing herewith. It will be noticed that the cylinders are very elementary in form. They consist of a steel fore and aft entablature into which the cast iron liners are dropped, the extension of the liner through the bottom of the entablature being covered by a welded plate jacket secured to the underside of the entablature, and made watertight at the bottom by a pair of rubber rings. The covers are shallow and of cast steel, the valve boxes being separate very much on the submarine lines. The gas joint between cover and valve box is a coned one, metal to metal, the water joint above it being a sliding one made with a pair of rubber rings. The reversing gear is rather out of the ordinary for Diesel engines, consisting of a pair of bell cranks for each valve, these bell cranks being mounted on oppositely disposed eccentrics on a fulcrum shaft. According to the position into which this fulcrum shaft is turned, one or other of the bell cranks engages with the cam below it, and thus gives the correct functions for ahead or astern running as the case may be.. From the bell crank which happens to be in action the motion is carried to the valve by means of a lever moved by the bell crank and connected by a pull rod to the lever on the valve box.

The fuel cams are mounted in sliding blocks on the cam shaft and while the fulcrum shaft is being turned through 180° these blocks are slid along the cam shaft, putting, say, the ahead cam out of action and bringing the astern fuel cam under the fuel valve tappet. A somewhat similar arrangement is used for reversing the air starting valves, which are arranged in one box from which pipes are led to the separate cylinders through a non-return valve at each cover. The gear is operated by an air ram, the movement being steadied by an oil ram acting as a dashpot.



Engines of the "Trefoil."

The cylinders are 17 in. diameter with a 27 in. stroke and the pistons are not water-cooled. They ran quite comfortably even under a considerable overload. In passing, I may say that in the engines of the *Marshal Soult*, with whose working I am well acquainted, there was considerable piston heating at times, in fact the lubricating oil on occasions caught fire though it was never necessary to stop the engines on account of defect throughout the war. This fact I see was mentioned in the discussion arising out of the paper referred to. The heating had nothing to do with the system of injection, however, being simply caused by the fact that as the vessel often had to sit on the mud, the suctions became choked and the engine did not get any circulating water till the ship had left her moorings and had run for a little time properly afloat. A steam ship, as we all know, would have corresponding trouble—with condensers for instance —in similar conditions. The remedy was to fit a special sea suction as was afterwards done and as is also done for steam ships in the same circumstances.

The *Trefoil* was delivered in May, 1917, and after a very satisfactory sea trial we proceeded to Liverpool, where from August, 1917, to October, 1918, we were engaged in and around Liverpool and Manchester docks and running up and down the Canal with occasional runs to Lough Swilly, Bantry Bay and the Clyde. Being a twin screw ship we relied upon our own manœuvring powers in the Docks and Canal to a greater extent than if we had been a single screw steam vessel, and, as may be imagined our reversing gear had a very thorough test, telegraph orders in the early days coming down sometimes in such quick succession that it was almost impossible to register them in the log. The ability of the oil-engined ship to go astern at full speed, almost immediately, and its possession of what might be regarded as a full head of steam, a full vacuum, and heavy fires at instant call, was fully brought home to me on many occasions and there is no doubt that in the canal and docks we carried out, unaided, many evolutions which would have resulted in piling up a steamship which had attempted them. We had no hitch at any time with the reversing gear and though we suffered from constant changes of junior engineers the new men soon picked up the running of the engines, and after a very short time were quite competent to handle the engines. Judging from my previous experience with air injection oil engines I can say that in the *Trefoil* the solid injection engines started, if anything, more rapidly and more smoothly than did the other type. When a number of engine reversals were required we started one or sometimes both of our air compressors to keep our starting reservoirs charged. As the compressors were not in use on other occasions they gave us no trouble, there being ample opportunity of examining and refitting the valves before they became troublesome. There is no doubt that the reversing of oil engines is a matter which has been satisfactorily settled in this, as in many other Diesel designs, and that the fears often felt in this connection are not now justified. It stands to reason that the gear must be kept in order, as in the case of any other apparatus, but in the *Trefoil's* engines the mechanism was easily understood and was readily accessible, so that a little intelligent interest was all that was required to keep the gear up to concert pitch.

The compressors were two in number, each three-stage, and were driven by Vickers' solid injection engines, the air being stored in six riveted air reservoirs. We found it was necessary to keep these reservoirs well drained, otherwise pitting began. The reservoirs were about 2 ft. 6 in. diameter by 11 ft. long, with a manhole door at the end, so it was quite easy to examine the interior and to keep them in good condition. This type appeals to me more than did the long narrow bottles used in some other ships I have seen, as I imagine that the insides of the latter are likely to suffer owing to the greater difficulty of inspection. The maximum pressure of air was 600 pounds per square inch, but everything was quite in order as long as we had a pressure of 250 pounds. We usually kept a reservoir charged, and shut off the main air line for use in starting our compressors in case of an emergency such as a leaky joint, but, as a matter of fact, we were always able to obtain plenty of air and had no need to fall back upon our reserve. The engines, as I said, started very quickly, especially when warm, and I can quite believe that the absence of the injection air, which must have a cooling effect upon the charge, accounts for the easy starting from cold. The only precaution to be taken is to prime the fuel system thoroughly with the hand pump, if the engine has not run for several days, or if the fuel piping has been disconnected for any purpose. If this is not done it is plain that considerable starting air may be used in driving the engine simply to pump the air out of the fuel system. I know of a case where owing to neglect of this elementary precaution, a ship twice ran out of air before the main engines were got going. This took place in an air injection installation where there is the liability of air locks occurring in the fuel piping, owing to the injection air being coupled to the fuel system.

On each of the *Trefoil* main engines there are two fuel pumps, each consisting of a two-throw pump driven by spur gearing from the cam shaft. The diameter of the plungers is ·75 in., and the stroke 1·5 in., the pump running at engine speed. The design of the pump is, broadly speaking, similar to that of the ordinary Diesel engines, though owing to fuel pressures of 4,000 pounds being used the parts are more massive. The eccentric straps in particular are very wide and are white metal lined.

Both suction and discharge valves of the fuel pumps are in duplicate, the two suction valves being almost in contact so that the suction valve control affected both almost together. A vent valve is fitted below the upper discharge valve. The output of the pump is varied by a hand lever which raises or lowers the fulcrum of a sway beam to each pump, one end of which beam is operated by the plunger motion, while the other, which is fitted with an adjustable tappet holds the suction valves off their seats for a longer or shorter portion of the stroke of the pump according to the position of the control lever.

The pumps discharge to a fuel main which is a pipe running the length of the engine and fitted with leads to each spray valve, the latter being a cam operated spring loaded valve, similar in general principle to the ordinary Diesel valve, though there are no pulveriser and flame plates or injection air connections. The fuel passes from the main to the spray valve and when the spray valve is opened by the fuel cam the fuel passes through the spraying nozzle to the cylinder. The nozzle is a simple affair consisting of a ball end in which are drilled five holes, each about 19-thousandths of an inch in diameter and countersunk on the outside. The nozzle is made of tool steel and stands up very well to its work, though eventually the holes become eroded by the passage of the fuel. The nozzles are small items, easily replaced, and as they last about 12 months, fitting a spare is a trifle. I know there was considerable experimenting carried out by the makers before the right proportions of the spraying gear were discovered, but from the sea-going point of view the whole system is almost ridiculously simple. All one had to do was to adjust the pump control lever to give the required pressure in the main and then it would remain constant for hours. Now and then a touch on the lever might be necessary to bring the pressure gauge pointer back to its exact position on this dial. When altering speed the opening of the spray valves was altered by a second lever and the pump control afterwards adjusted if necessary.

The spray valve is, of course, the soul of the job, as it determines the fuel admitted and it must, therefore, be kept perfectly tight. This, however, is easily done and all that is required is care and an interest in the job. We made it a practice to touch up the valves about every month and to give the valve a rub into the seat using a little crocus powder or rotten stone. After wards the valve was fitted to a hand pump and tested to full pressure to see it was tight and that when the lever was tapped by a mallet the spray from each hole was clean and that the valve shut off sharply and without dribble.

Our valves had no external gland on the spindle as there was a bell crank inside the valve box for lifting the spindle and the gland was on the bell crank shaft which turned in the gland—the fuel load on the bell crank being taken up by a ball race inside the gland. The glands of the spray valves and of the pumps are packed with soft packing. We generally used Palmetto packing with a turn of Parrametto packing, which is a bit harder, or a leather ring at the top and bottom of the stuffing box, but most soft packings will do. The Tuck's type of packing may be used as long as there is no indiarubber core which would be affected by the fuel. It is not so much a matter of the material of the packing as the care with which it is fitted in the stuffing box. If the stuff is pressed evenly into the box throughout the operation the gland may be slacked back to hand tight after squeezing the lot home and the rods or plungers will run sweetly and soon get a beautiful polish on them. If there is any leakage it is best to repack the gland at the first opportunity, and I always forbade any undue heaving up as it simply tended to score the rod and in the case of a spray valve to prevent rapid closing which would result in smoke and waste of fuel. In my opinion much trouble would be saved with rods of all descriptions, especially in the water ends of bilge pumps, if a similar routine was followed.

The fuel pressure of 4,000 pounds to the square inch might cause misgivings to engineers unused to such pressures, but such are quite unfounded and there is no difficulty at all with the joints which are designed for the pressure, or with the glands. I found no more trouble with the solid injection pump or the fuel service than I had with the corresponding fittings in air injection engines I have served with afloat. Perhaps I have been unfortunate in my experience with the latter type of engine, but I must confess I came to regard high pressure air in connection with the fuel in a Diesel engine as a dangerous shipmate. Diesel experts would doubtless have something to say in the matter, but a plain sea-going engineer like myself will probably be excused by most members of this Institute for feeling a bit uneasy in the neighbourhood of an engine that had on occasions, in my presence, burst cylinder covers through some vagary of the injection apparatus. With the pump injection, on the other hand, everything seems such plain sailing and one has confidence that smoke is probably the most serious result of any failure to treat the system in the proper manner.

I believe that the modern air compressors are much better than those in use some years ago, but even so I reckon that they form the weak spot in the ordinary Diesel engine. Apart from valve trouble the pipes are apt to be scoured and oxidised by the hot compressed air, and I see that duplicate compressors are fitted in a number of engines. Explosions in the intercoolers due to the wrong sort of lubricating oil, or as far as that goes, whatever oil is used, seem to take place every now and then in land stations, so I am inclined to think that in ships where conditions are not so favourable, air injection should be regarded as a half-way house to something better.

Speaking of intercooler explosions, I remember that while engines were being built I was talking over compressor troubles with the makers, and was at first surprised at the statement that as the result of one of these explosions a steel plate separator, though proved to be of perfect quality, had shattered as if made of some very brittle material and that a needle of steel had been shot into the breast of one of the engineers with fatal results. The cause of this must evidently have been something in the nature of an explosion wave or else a detonation as opposed to the less sudden pressure rise described as an explosion, and owing to oil being very finely broken up, mechanically or perhaps even chemically, and in close association with the oxygen in the highly compressed air. I have thought that the heavy knocking sometimes experienced in air injection engines, when the blast air was excessive, might be due to something similar and that the bursting of covers and the fracture of crankshafts, sometimes happening in engines with the air system, might be due to the same sort of thing in a less violent form. However that may be, nothing of the kind as far as my experience goes can happen with a solid injection engine. High pressures can be obtained if the spray valve leaks or if injection is set too early, but these are matters which require no particular knowledge or care to avoid. Even so the pressures seem to be more gently imposed and in the submarine engines which, by the way, have no cylinder relief valves, I have seen a spray valve held continuously open for demonstration purposes with no ill effects, the result simply being smoke and heavy knocking or "tonking" in the cylinder. The indicator diagram is, of course, very peaky in such circumstances, and I would prefer not to carry out "stunts" of this sort on the larger engines in my own ship, but I feel sure that the results of abnormal conditions with solid injection are more gentle than with air injection, however the indicators may fail to distinguish between the two. It was certainly remarkable how easily the *Trefoil's* engines picked up even at low speed with a practical absence of shock, the whole performance reminding one of our old friend, the steam engine, though the oil engine could pick up full power or a bit over in a fraction of the time taken by the steam engine.

During the discussion on the former paper I saw that the solid injection set was stated as not likely to prove a commercial success, except in the smaller sizes, owing to an absence of turbulence. I don't profess to know what happens inside the *Trefoil's* cylinder, and as a sea-going engineer I am more concerned with fuel consumption per mile run or per shaft horsepower. The Vickers' engineers claim that the solid injection system will give at least as good consumption as air injection in similar circumstances, and judging from the *Trefoil's* running which took place on any service fuel that we happened to be carrying for the fleet, I see no reason to doubt their claim. As regards consumption, the unwary shipowner and engineer are sometimes misled by figures being given per indicated horsepower, or sometimes simply per horse-power. With steam engines of more or less standard efficiencies this is a good enough basis of comparison, but it would be better to bring all consumptions to a brake or shaft horse power basis, as is done for turbines. This eliminates the frequent error in indicator diagrams, which, as ordinarily taken, are none too reliable on oil engines where temperatures are high, and the figures would then have a practical meaning. The indicated power consumption gives the air injection engine an unfair advantage, as with equal effective consumptions the more inefficient the engine is mechanically, the better the consumption appears for comparison.

In commercial work the mean pressures are moderate, so even if air injection permits of higher duties these will not be required for mercantile marine work. There is in the cylinder at ordinary ratings about twice the amount of air theoretically required for combustion, so, as we have often found in the *Trefoil*, there is room for a fair overload without needing any extra air from a blast compressor. If the engine is in good condition the exhaust remains practically clear even under overload.

Smoke in a Diesel engine seems always to be one of the first things to be criticised in my experience, but I notice that a good cloud is regarded as rather impressive on a steam trial of a coal fired ship, while when burning oil under boilers it is preferred for reasons of economy to maintain a slight colour at the funnel. Semi-Diesels give considerable smoke on service in many circumstances without doing any harm, and it has occurred to me that it would be wise to regard smoke in a Diesel rather as a matter for investigation than for immediate censure. I say this because I have found that while with solid injection any fault is soon evidenced by smoke, yet the exhaust pipes remained practically clean. On the other hand, with air injection engines the exhaust might be very good yet considerable deposit be found in the pipes and on the valves. I can imagine various possible explanations for this, but as I have no means of testing the truth of my idea I simply put it forward as a fact which may be of interest, and one which I think has not been previously mentioned.

The lubrication of the *Trefoil's* engines was forced throughout. These low headroom engines had no division plate between the crankcase and the cylinders such as a purely commercial engine would have, so that the lubricating oil was contaminated by any dirty or burnt oil falling from the pistons, as in a trunk engine. In addition, the bed-plate and drain arrangements led to bilge water occasionally getting into the oil. Even so there was never any thought of bearing trouble, and I am convinced that if only steam marine engines were fitted with forced lubrication to the principal parts, the wear and tear and final expense would be immensely reduced. It is a common enough practice nowadays with auxiliary engines, and I am told that in some of the last reciprocating-engined battleships in the British Navy it was fitted to the main engines with most satisfactory results. It is, of course, necessary to prevent salt water mixing with the bearing oil, but this presents no difficulty in a job designed for forced lubrication. When likely to remain in harbour for any length of time I made it a practice in the *Trefoil* to use the hand lubricating pump to pump some fresh oil through the bearings after stopping. This washed out any water which, if allowed to remain, would pit the journals.

A point that is sometimes overlooked with oil engines is the necessity of periodically cleaning out the water space. What is a matter of routine in the case of a boiler or of a condenser is apt to be regarded as an irksome task in an oil engine. It stands to reason that just as furnace crowns will come down if boilers are allowed to get dirty, so covers may crack or pistons seize if the water spaces of an oil engine are allowed to fill with mud. Nothing should be allowed to stand in the way of a periodical swilling out of these spaces. The oil engine has come to stay and the routine must be made to suit its requirements.

In drawing my remarks to a close it may be interesting to those unacquainted with oil engines to give some comments upon the oil engine from the personnel point of view. Among the number of untrained men who have passed through the engine-room of the *Trefoil* I found the majority on arrival not in favour of the oil engine. They did not seem confident in the machinery nor in themselves, feeling that they lacked intuition to go to the source of any trouble if it arose. After a while they learnt that a good oil engine does not require a professor to run it, and that it may be run with less trouble than a steam job. They then became enthusiasts for the oil engine as a self-running apparatus, and it was necessary to keep them alive to the fact that there remained work for them in the way of inspection even if repair was not needed. Their general expression of opinion after settling down was that the oil-engined ship was far preferable to the steamer, and I am convinced that from the *Trefoil* there has gone out a number of men intent on spreading far and wide the advantages of the solid injection oil engine. You will have gathered my own opinion from the notes I have given, and I can honestly say that my log shows much more work was occupied in keeping steam auxiliaries in order than was devoted to the main engines, experimental and special in design though they were. In my judgment, given a little instruction in the construction and working of the machinery there is nothing to prevent any average steam engineer with some power of adaptability from becoming in a short time fully competent to run an oil engine of large power, and no fear that any highly scientific knowledge is required need be entertained.

If engineers can be made to hold these opinions, the fuel consumption, which is well below half that of oil-fired steamships, should convince owners that it is worth while to consider oil engines for ships for which engines of the power required are being offered, and I am of the opinion that before long we shall see few ocean going cargo ships built with steam engines, and that when freight costs fall, the economy of the oil engine will give to the owner of the oil-engined ship an unassailable advantage over his less enterprising competitor.

As experience is increased it will doubtless be found that there is room for a variety of makes of engine. Judging from the result of development of the steam engine which in its early days took many peculiar shapes, advantageous on paper, but eliminated as the result of sea experience, I am inclined to think that even allowing for the difference in the driving power, evolution will for ordinary commercial purposes result in oil engines being arranged as regards their principal parts, very much on the lines of the latest reciprocating steam engine. When I find myself being persuaded of the virtues of some very novel arrangement of an oil engine, I often realise that the same arguments would apply to a steam engine on the same lines and I become doubtful if the unorthodox type will receive more than a passing favour for ordinary cargo ship use.

As regards the choice between the four and two-stroke cycles for medium powers and speeds there seems little to choose on the grounds of weight and space or cost if the same rating is followed in each case. This being so, I favour the four-stroke engine as being practically just as simple as the two-stroke engine, the absence of the scavenging pumps being a great advantage, and subjected to a less continuous high temperature.

The final method of working auxiliaries will, in my opinion, be electric, unless tank heating or other special circumstances cause steam boilers to be in use, but doubtless many owners will be wise in retaining steam auxiliaries till the oil engine has become more familiar to their engineers, thus introducing one novelty at a time. Electric auxiliaries for the engine, such as circulating and oil pumps, are in my opinion much preferable to the pumps driven by sway beams off the engine, and will ultimately become universal.

Solid injection, whatever may be advanced against it from the theoretical side, is undoubtedly a working proposition, and I expect to see it taken up more by various makers now that it is coming into use in commercial vessels and its wide and successful use, hitherto more or less secret, is becoming known. The Americans, among other foreign countries, seem very much alive to its possibilities and it would be well, now that Great Britain is seriously taking up marine oil engine manufacture, that her engineers should give it full and unbiassed consideration.

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ADDENDA.

The following is a description of the engines and some notes regarding the overhaul which may be of interest:—

The engines of the *Trefoil* are of the "Vickers" heavy oil type, operating on the Diesel principle and adapted for the combustion of heavy mineral oils of high specific gravity and flash point. They are of the vertical four-stroke cycle, single acting type, with trunk pistons and enclosed crank chambers.

The impulse cylinders are three in number $10\frac{3}{4}$ inches diameter and 13 inches stroke designed for a speed of about 320 revolutions per minute. They are made with separate liners and jackets, both parts being of plain design facilitating rapid removal, if necessary.

The Cylinder Liners are made of special cast iron, machined all over and so mounted in the entablature as to be free to expand.

The Cylinder Jackets are made of sheet steel and are so constructed and attached as to be readily removable.

The Cylinder Covers are made of cast steel of strong construction and design. They are attached to the entablature and plate steel columns, the arrangements being such that the impulse loads in the cylinders are transmitted from the covers through the columns to the bedplate without passing through the cylinder bodies. Special care has been taken in arranging for the most efficient cooling of the covers. In each cover are fitted one induction, one exhaust, one air starting and one relief valve, and also one spraying nozzle.

The Connecting Rods are of forged steel and are coupled direct to gudgeon pins carried in the pistons.

The Bedplate consists of strong cast steel girders tied together longitudinally by wrought steel channels, the whole being mounted on a strong cast iron seat, to which they are secured, and which acts as an oil well for the forced lubrication system.

The Crank Shaft is made of forged steel, the cranks being arranged in the sequence, and with the necessary balance weights to produce the best possible combination of balance and twisting moment of the set.

The Injection of the Fuel Oil is accomplished by means of the "Vickers" special patented process in which the oil is sprayed directly into the cylinder without the use of blast air. A fuel oil pump of ample capacity, driven by gearing from the engine, is provided, discharging, at a pressure of from 3,000 lbs. to 4,000 lbs. per square inch, into a common pipe or rail. A mechanically operated valve is fitted on each impulse cylinder to measure the fuel oil supply; a pipe is led from the rail to each of these valves and thence to the spraying nozzle in the cylinder cover. A number of very minute holes are bored in the nozzles which, by means of the high pressure of the oil discharge, split the fuel oil up into the fine spray necessary for complete combustion. Screw down valves are fitted in connection with the spray nozzles by means of which any individual cylinder can be cut out.

Forced Lubrication System. A complete system of forced lubrication is provided, both for the cylinders and for all the more important bearings. A pump driven by the engine is fitted for each of these services. For the bearing oil the pump draws from a drain and settling tank, the oil being collected in the cast iron seat and returned to this tank to be used over again after settling and passing through the strainers. For the cylinder lubrication a multiple plunger pump is provided having an independent plunger for each cylinder. Adjustable sight feeds are fitted on the suction side of each plunger for regulating the supply of oil, and in addition a hand pump is fitted in the ship by means of which any cylinder can be flushed with oil at any time. A hand pump is also provided for priming up the lubricating system before starting the engine.

A Circulating Water Pump is provided, driven by the engine, for supplying the cooling water to jackets and other parts where it is required.

A Hartnell Governor is provided to keep the engine speed constant under the varying loads experienced in charging the air storage reservoirs from atmospheric pressure up to 600 lbs. gauge pressure.

As the engine control is most conveniently effected by regulating the lift of the fuel measuring valve, and as a considerable amount of power is required to perform this operation, the governor is arranged to carry out the movements through the medium of a servo-motor. This servo-motor consists of a cylinder containing a piston which is coupled through link work and levers to the fulcrum shaft carrying the measuring valve operating levers. Oil under pressure from a pump is admitted to either side of the servo-motor piston, the only function of the governor being to control the admission of oil to the servo-motor. This is done by means of a balanced piston valve which is easily moved by the governor.

An Inertia Governor is provided, the function of which is to stop the engine should the speed increase 20 per cent. above the revolutions required for full output, due to a sudden removal of the load, or failure of the "Hartnell" governor to come into operation quickly enough. This governor acts by releasing the oil pressure from the rail.

A Flywheel is fitted to the engine to assist in obtaining smooth running of the set; it is of cast iron and also acts as a turning wheel.

Air Compressor.—The air compression is performed in three stages, there are two first stage cylinders each 11·15 inches in diameter, one second stage cylinder 7·2 inches in diameter and one third stage cylinder 3·9 inches in diameter, the stroke in all cases being 12 inches. The compressor is coupled direct to the engine and has two cranks, the cylinders being arranged in tandem; one first stage cylinder and the second stage cylinder are operated by one crank, and one first stage cylinder and the third stage cylinder are operated by the other crank, the first stage being the lower cylinder in each case.

The Cylinders and Jackets are of special cast iron in all cases, the jackets being cast with the cylinders. In the first stage, inlet ports are fitted—in addition to the inlet valve at top of stroke, these ports are fitted with a slide by means of which the amount of air admitted can be regulated.

The pistons, two in number, are made of special cast iron, the first and second stage pistons from one casting, and the first and third stage from the other casting. The gudgeon pins are fitted in the first stage pistons.

Cooling water is supplied to the jackets, covers and other parts by the engine pump. The air between the stages is cooled by means of water jacketed pipes for first and second stage discharges, separators being fitted in these pipes for the purpose of draining any accumulation of water. The air from the third stage discharge, in addition to a water jacketed pipe, passes also through a separator and coil cooler before being finally admitted to the reservoirs.

While overhauling the main engines the following notes were recorded for future guidance:—

Cylinder Covers.—Considerable difficulty was experienced in the lifting of the main engine cylinder covers, it being found necessary to remove several heavy exhaust pipes before covers could be lifted. To avoid this in future, two methods were submitted:—

(*a*) To substitute new branch pipes and a distance piece which could be readily disconnected.

(*b*) Alter the exhaust pipe at the expansion joint to give sufficient space to allow the cover to lift clear and have a distance piece fitted between the valve and original branch pipe.

However, neither of these two methods is being adopted ; the first would cost too much and the second would give too little room behind the main engines, but the glands in the exhaust pipe branch piece are being fitted in halves, so that the packing may be withdrawn, and with the exhaust valve out, the branch piece can be turned round and the cover lifted clear.

Valves.—On removing the valves from the covers, all the valve faces were found in good condition, but the internal tubes in several of the exhaust valves were perforated. All exhaust valve tubes required renewal.

Pistons, Rings and Liners.—When the cylinder covers were removed the absence of carbon was very noticeable. The first two piston rings and in some cases the first three, were stuck; the remaining rings, however, were working satisfactorily.

All the cylinder liners were in excellent condition.

Water Jackets.—On removing the liner water jackets they were found with mud accumulated below the inlet and banked up opposite.

Mud doors have now been fitted to admit of periodic cleaning.

Crankshaft and Main Bearings.—On the crank pin there was found to be a wear of 5/1000in., but the surfaces of all bearings were very good, Both crankshafts were re-bedded and all top and bottom ends adjusted.

Turning Gear.—The frame of the starboard turning gear required to be refitted to engine sole plate. On the port engine it was found that $\frac{1}{2}$in. required to be taken off the worm on the vertical shaft, to allow it to clear the crank shaft wheel in all positions, while out of gear.

Cooling Water Pumps.—When these were opened up the valves, spindles and springs were found badly worn, but with a lighter type of valve being fitted, no further trouble is looked for in this direction.

Air Compressors.—The power cylinders, pistons, rings and valves were all in good condition. The connecting rods were also in good condition with the exception of two on starboard engine, which were bent at the neck and being replaced from spare. (Water in the cylinder when starting up may have been the cause of the bent rods.)

On the crank pin there is a wear of 5/1000in., most of which has taken place on the underside of the pin, probably due to the engine's high speed and the inertia of the piston on the exhaust stroke. The surfaces of all bearings were very good.

A number of the rings in the port air cylinder were stuck, caused by a salt water leak in the first stage cooler; this leak will be made good and the rings renewed. The rings in the starboard air cylinders were all in good condition, but as considerable wear has taken place on the first stage rings these require renewal.

Control Gear.—The fuel pressure pump suction valves having a constant lift, considerable difficulty was experienced in maintaining a suitable pressure for all loads. The only control over the pump was a throttle on the pump suction pipe, but as this was operated off the spray valve lift control, increasing or decreasing the fuel pressure meant altering the lift of the spray valve, which is not desired at all loads. In this type of pump it is not possible to vary the lift of the suction valves and to avoid the expense of new pumps, a hand controlled throttle on the suction pipe is being fitted in addition, and with this fitted, better results are expected.

Cooling Water Pumps.—On opening up these pumps it was found necessary to have them re-bored; this, however, brought bad metal to the surface, so new chambers were cast.

AUXILIARY MACHINERY OVERHAUL.

Dynamo Engines (two).—On opening up these engines the L.P. cylinders were found badly worn, requiring to be re-bored and new pistons and rings fitted, the bearings also being readjusted.

Fire and Bilge Pumps (*two*).—Both pumps were in good condition, but new steam rings, also new bucket rings in port pump were required.

Oil Fuel Pumps (*two*).—Both pumps, rods skimmed up and new rings fitted.

Boiler Room Fans (*two*).—Piston rods skimmed up, starboard cylinder re-bored, new rings fitted for both, and bearings adjusted.

Pump Room Fans (*two*).—Opened up for examination and found in good condition.

Engine Room Fan.—New rings fitted and bearings adjusted.

Feed Pumps (*two*).—New steam and bucket rings fitted, also valves lined up.

Air Pump.—New steam rings fitted. Bucket valves cleaned and turned.

Circulating Pump.—New bushes fitted on impeller shaft. New steam rings fitted, and bearings adjusted.

Condenser.—Cleaned, new protection plates fitted, tested and found satisfactory.

Evaporating Plant.—Pump sent to shop and overhauled. Coils annealed and valves ground in.

Cargo Pumps (*two*).—Valve seats being reseated and valves ground in. Pistons and shuttle valves and buckets found in good condition.

Ballast Pump.—Water end being re-bored and new bucket rings fitted.

Capstan and Windlass.—Bearings adjusted.

Steering Engine.—Stronger bedplate fitted; piston valves and steam rings renewed, bearings adjusted.

Boilers, Water Tube (*two*).—Fire row to be renewed, all other tubes and drums in good condition.

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The Chairman: The feature of Solid Fuel Injection is doing away with the complication of Air Compressors as required in the original Diesel Type Engine, and although a number of vessels installed with engines of the type are giving satisfaction on long voyages, it is evident that a process of simplification is in being among British engineers such as took place on the introduction of the Compound Steam Engine. From all I hear, the working and maintenance of large Marine Oil Engines is readily acquired by Marine Engineers, and some ships are running successfully with engineers transferred from steam vessels to oil-engined vessels in the same company.

A Member: The paper under discussion represents a whole mass of common sense, and from that point of view it is a most valuable contribution to the papers which have been read before this Institute on this subject.

If there are any manufacturers of air compressors here tonight they will probably feel somewhat anxious with regard to the future prospects of this branch of industry. At the same time there appears to be a good opening for these makers as marine engineers, because the paper has clearly shown that the marine oil engine may be run with very little knowledge of this class of engine.

Although the paper is discussed under the heading of "The Solid Injection Engine," I would personally object to the use of such a term. One has always heard the words "solid injection" coupled with "solid exhaust" and as the nomenclature of the oil engine generally has suffered very largely, there is some justification, I think, to use the term "mechanical injection," which after all is a more correct technical interpretation of the system under discussion.

Going back to the last paragraph of the paper, the Author states that "the solid injection, whatever may be advanced against it from the theoretical side, is undoubtedly a working proposition." It is not intended to discuss the theoretical aspect of the problem, but I would like to state that from a theoretical point of view the mechanical injection engines can be proved to be more efficient than the air injection type.

Examining for a moment an ordinary diagram (Figure 1) of a heavy oil engine we can refer to point A on the diagram as the cut-off point similar to the cut-off point in the ordinary steam engine. It can be proved that the thermal efficiency increases with a smaller cut-off ratio and an actual experiment gives the following results:—

Load.	Full Load.	¾ Load.	½ Load.	¼ Load.
Diagram	A	B	C	D
Cut-off Ratio from indicator diagram	2·45	2·11	1·56	1·39
Thermal Efficiency	51·4%	52·4%	55·75%	60%

Owing to the absence of the cooling effect of the expanding injection air and owing to the consequent smaller amount of heat to burn the fuel, the cut-off ratio in a mechanical injection engine is lower, with a corresponding increasing thermal efficiency.

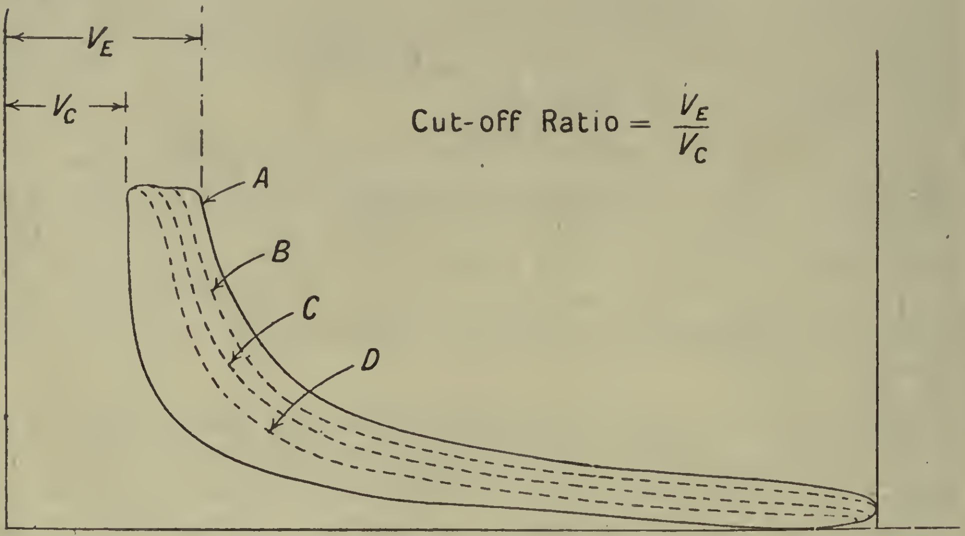


Fig. 1.

From a technical point of view we have to deal with the question of atomisation and turbulence. It is stated as a fact that the expanding air increases in volume something like 75 times its original volume, thereby producing the atomising effect. However, it should be remembered that the oil on being heated expands, and the relative volume of a globule of oil in the solid and gaseous state is something like 160. Although it is recognised that in the available time the oil cannot be gasified completely, the available temperature is sufficient to gasify a proportionate volume of oil to give approximately the same increase in volume as the expanding air, hence a similar atomising effect can be produced.

In my discussion on the previous paper, I went into the question of heat transference from the heated air compression to the oil globule and there is, therefore, no necessity to go into this point again. There is, however, no doubt that the required amount of atomisation and accompanying turbulence may be produced equally efficiently with mechanical injection as by the air injection system. The question is then asked as to why with equally good combustion the fuel consumption in a mechanical injection engine is not lower in proportion to the increased mechanical efficiency between the two types of engines, which owing to the absence of the air compressor is improved from 75 per cent. to about 83 per cent. Whilst it is admitted that in certain instances the mechanical injection engine has not reached the same fuel consumption as the air injection engine it has been proved that under certain conditions the mechanical injection engine can deal with heavier, *i.e.*, inferior, fuels than the air injection engine. We should remember that after Ackroyd Stuart recommended the mechanical injection system, Diesel recommended the use of the air injection system which was adopted universally. Continental and other manufacturers who tried the mechanical injection method condemned it as a failure, and not until the British oil engine industry took up the matter seriously has it been proved that there is a practical solution to this important system. The possibilities of using a valveless engine (either simple or compounded) offer a great inducement to us to study this important system. Its construction will be simple, thereby increasing its reliability and facilitating easy manipulation on the part of the personnel.

I am sorry the Author did not touch on the question of fuels. He gives no indication of what fuels he has been employing. I would suggest that such information is invaluable in order to make a complete survey of this important problem. The marine engineer can give most instructive information to the designer and manufacturer by making notes and records of his practical experience. Only by such co-operation will it be possible to design an engine which will be equal, if not superior, to the general behaviour of the modern steam engine.

There are in this country under construction five designs of mechanical injection engines for stationary purposes and three for marine purposes. I believe that in Germany there are three manufacturers actively engaged to-day, whilst in America we have five further manufacturers who all believe in the ultimate success of the mechanical injection engine. Whilst these manufacturers represent a very small nucleus of the total firms engaged in this industry, their efforts indicate the tendency of design and it is our duty, not only from a commercial but from a national point of view, to fully investigate all the problems—of which there are a great many — connected with the mechanical injection engine in order that manufacturers in this country may be able to compete successfully with the efforts which are being made, and which will continue to be made, very vigorously in other industrial countries.

A Visitor: My experience is not that of the Author of the paper when he says his reversing gear generally works. Mine generally does not. The paper is of great interest and has obviously been written by a practical engineer who has much more interest in the £ s. d. side of affairs than in the scientific. He does not know exactly what goes on in the cylinders and really he does not care. But, on the other hand, we who are interested in the development necessarily should care and endeavour to find out what it is that takes place inside the cylinders. The Author does not mention the type of fuel employed.

The situation in the world to-day is that the amount of oil available for fuel purposes, though large, is not growing as rapidly as the demand, and unfortunately the Diesel type of engine cannot use a very considerable proportion of the new oils turned out from the freshly opened fields in various parts of the world. The particular oils which the Diesel can use are comparatively limited.

It is unfortunate that in the efforts to use these heavy oils Diesel engine makers have had to employ blast air at a high pressure in order to sufficiently atomise the lower grade fuel.

They have had to employ a very high compression in order to overcome the inherent difficulty introduced by the expansion and consequent cooling of the blast air which unfortunately brings down the temperature at the very point where ignition has to commence. This necessitates a compression which is greatly higher than that theoretically necessary to bring about spontaneous ignition of the fuel. What we should like to know regarding the solid injection engine, or better called the mechanical injection engine, is whether fuel, injected though it be at pressures up to 7,000 lbs. per square inch, is really atomised or whether it proceeds across the cylinder in the form of continuous needles of liquid fuel. This makes a very great difference. The more efficiently a fuel can be atomised the more rapidly it will burn. The previous speaker's explanation of how mechanical injection brings about that degree of pulverisation which is in some degree commensurate with that brought about by air injection is very interesting and would be valuable material for use by gentlemen engaged in selling mechanical injection engines. Unfortunately he gives no proof that the injected fuel is atomised, and if it exists as a needle of liquid it has to be vaporised and the rate of combustion is largely dependent upon the rate of diffusion of the air into the gas envelope which occupies the place of the thread of oil ejected from the nozzle. If you can get a spray, the surface exposed is much greater than is exposed by the same weight of oil in the form of a cylindrical thread. It is interesting to note that if two vaporisers or atomisers give particles having diameters in the ratio of 1 : 2, under the same conditions of compression and temperature the chances of the oil being burnt in the two cases are in the ratio of 1 : 16. Fine pulverisation of the fuel diminishes the probability of the formation of hot spots on the cylinder head or walls or combustion top. I am absolutely in agreement with the previous speaker in considering that the future is in the hands of those who go in for solid injection. It is right in principle, and it is only a question of mechanical adaptation that will enable us to achieve results by it. In fact, I think they have been already achieved. What is required is an engine that will be capable of using the widest range of oil, which shall have the highest thermal efficiency and be capable of allowing increase of cylinder size. If the temperature in the interior of the cylinder is too high the skin surface of the steel or iron is raised to a very high temperature, and before the heat can be carried away the expansion of the top layer of iron or steel has been carried beyond the modulus of elasticity and the iron or steel becomes covered with extremely minute cracks. This was found to be the case in an engine built by Junkers. That apparently limits the temperature that can be attained in the interior of the cylinder. That is the view entertained by Junkers.

The Chairman: I think the initial and upkeep costs are two important features. and any information on this point will be of value, because a good many people are enquiring as to the cost of upkeep and the time required for repairs in port. You have to consider all these things in regard to the value you are going to receive in service. I have also heard of requirements for a standard Diesel Engine fuel oil, as there are great differences in the characteristics of available fuels; in this it is reasonable that the engine that can work on the widest range with minimum adjustment will be the type that will find favour for marine propulsion.

A Member: Referring to explosions of intercoolers on air compressors, I have found from experience that the water-cooled copper coil type gave considerable trouble owing to the large second stage coils (20 atmos) vibrating against the sides of the water jacket, thus causing a wasting and leakage of the coil. After an explosion of a third stage coil (60 atmos) the remainder of the coil was examined, and it was found that the hot compressed air had severely scoured the internal wall furthest from centre of the coil, down to a dangerous thickness; owing to the above scouring effect on the coils, it was found essential to frequently remove them for a weight test.

A Visitor: The Author states they were carrying oil for the Government. That would mean oil within the Government specifications for a fuel oil. These are rigid, and it tells us at least that it was oil having properties within certain limit, and that would certainly exclude oil of the worst type.

A Member: I share the opinions expressed by previous speakers that the class of oil should have been mentioned in the paper, and I also think we should have the oil consumption of the *Trefoil*. The Author states the bunker capacity and the displacement, but he does not give the oil consumed per brake horse-power, and this is required if comparisons are to be made.

The Author mentions in the paper that the engines are not commercial ones, but that the *Trefoil's* engines were designed for a special purpose. I think it would have been of greater interest to us if the 1,500 B.H.P. had been obtained on less cylinders. In this ship it required 16 cylinders to develop the power which, in other ships, is obtained from six cylinders of larger dimensions, and consequently the mean pressure in each cylinder must have been low and not comparable with the mean pressures of the latest marine Diesel four-stroke engines fitted with air blast injection, as the latter have a mean pressure of from 80 to 90 lbs. per square inch. It is therefore not commercially sound that so many cylinders should be fitted to obtain such a comparatively small horse-power because the cost and weight per horse-power must be high.

I agree with the Author that rotary pumps will be more extensively used in the future owing to their being able to do their work with less overhauling and stoppages for repairs, and where internal combustion engines are used for the generation of electricity the motor driven rotary pump is in my opinion the best type of pump to use for all general services where there is a constant stream of water or oil to deal with.

In a Diesel-engined ship I think it is best to drive all the auxiliaries electrically, and we have found that with a Diesel engine driving a dynamo which supplies current to electrically driven winches that one ton of oil is equal to about seven tons of coal. There is a big loss of heat from steam winches and also from the piping connecting these to the boiler.

In regard to Engineers for motor ships, my experience is that those previously accustomed to steam engines soon become expert at handling internal combustion engines because they are interested in their profession and in new engineering problems generally.

A Member: The Author states: "The Vickers' engineers claim that the solid injection system will give at least as good consumption as air injection in similar circumstances, and judging from the *Trefoil's* running which took place on any service fuel that we happend to be carrying for the fleet." He has evidently struck the right chord and I very much welcome this contribution based upon his experience, and give credit to him, as we do to the early pioneers who went through this world in the early days in the steam-engined ship, from the *Savannah*, in 1819, onward. He also speaks of bell cranks. When I noted this, it looked to me as if there might be a possibility of slightly more wear and tear in one set than in another, and it occurred to me whether these are arranged to work automatically to reverse full ahead to full astern in the sixteen cylinders. There is also a reference not quite clear to me where the gear is operated by an air ram, so also as to the oil ram dash pot. The question arises also whether there is a buffer fitted, as we have these on some gears in order not to act too quickly. The pistons are stated not to be water cooled, and I gather that these engines are not for commercial service. The heat of the cylinders becomes very great, I understand, under certain conditions, and it would be interesting to know what causes this, as it sometimes leads to disaster. Is it due to a very thin volume of oil with a gas generated at the walls of the cylinder, or is there any acid action set up from the effect of the condensation and a small portion of oil that may be mixed with it?

A Visitor from the Bureau of Standards, Washington, D.C.: There is a Diesel engine of rather unique design now being developed in the United States, a comment on which may be of interest. It is being developed by Mr. Elmer J. Sperry, the inventor of the Sperry gyroscopic compass, as a compound expansion engine. The value of compound expansion in Diesel engines is clearly shown by the indicator card drawn on the blackboard, where almost seven-tenths of the working stroke is at relatively low pressure and the remaining three-tenths at really high pressure. Metal used in strengthening the large size of cylinder required for the low pressure expansion is not needed when the high pressure expansion is carried out in a small cylinder suitable for high pressure expansion. Mr. Sperry has designed and constructed two experimental engines developing about 500 horse-power each, running at about 800 r.p.m. and weighing from 15 to 17 pounds per horse-power, a very material reduction of present practice in Diesel engine design. The great difficulty which has previously been experienced in the compound expansion internal combustion engine has been the design of the transfer valve. The present engines consist of three cylinders on a single crank shaft, a high pressure expansion cylinder, a low pressure expansion cylinder and a compressor cylinder. After the charge in the high pressure cylinder ignites, it expands to a relatively low pressure and then is transferred to the low pressure cylinder through the transfer valve of barrel type, in which it is finally expanded to atmospheric pressure. Immediately after the high pressure cylinder has discharged into the low pressure cylinder the transfer valve is rotated through a small arc and the compression cylinder discharges its air through the transfer valve into the high pressure cylinder for further compression and ignition. The work of the compressor cylinder is therefore recovered in the subsequent expansion in the low pressure cylinder. The thermal efficiency obtained by Mr. Sperry with this engine is from 35 per cent. to 39 per cent.

A Member: The Author speaks very highly of the way in which his engines reversed. I am of the opinion it is usually more trouble to get a Diesel engine to go astern after running full ahead with way still on the ship, than it is to reverse a steam engine under the same conditions; for the reason that you have not the same power for starting the Diesel engine. Usually only some of the cylinders are fitted with starting valves; those cylinders have to overcome the pressure on the propeller before it can be started to revolve in the opposite way against the direction in which the ship is going. The Author states that if he had 250 lbs. of air it was quite sufficient. It would be interesting to know if all eight cylinders were fitted with air starting valves. If so he had more power than it is usual to fit on the majority of Diesel engines. I have experienced that it is impossible to get a Diesel engine to go astern while the ship is going very nearly full speed ahead, and that the ship must be allowed to slow down herself, otherwise you might use all your air trying to get the engines astern, and then not manage it, with the result that you have not sufficient air to start the engine when the ship has slowed down to a speed at which it is possible to revolve the propeller in the opposite direction.

A Member: A further point that occurs to me in connection with these oil engines is this, that in its simplest form, as a land engine, working continuously in one direction, it usually has an enormous amount of working parts. When we come to the point that as a marine engine it has to work either way, either ahead or astern, it is certain that these parts must be duplicated in many instances. With this in one's mind, and recalling the results that have been obtained by the electrically driven ships of the United States Navy, is it not reasonable to anticipate that in the near future, knowing as we do that the auxiliaries must be electric or steam driven in some way, that a Diesel engine or other oil engine in the ship, running continuously in one direction and generating electric power, is likely to be more the motive power of the future than even an oil engine for the main shaft, and some oil engine or other motive power generating electric current for the deck machinery and the auxiliaries?

The Chairman: In regard to auxiliary machinery being worked by electricity, that is the case in many large Diesel-engined ships. The Diesel electric generator engine responds to requirements of power required for either deck winches, heating or lighting by accurate automatic governing.

A Member: There is a point in connection with this engine which I think should be carefully studied by any land type air injection Diesel engine builders who are contemplating building marine engines.

In the engine here described it will be seen that the r.p.m. are low, and this allows of an increased propulsive efficiency.

With the air injection Diesel engine for land work it is an advantage to have the stroke more nearly equal to the bore, as the fuel burns more quickly, and parts may be reduced in weight, thus the revolutions tend to be greater, which is a disadvantage in marine work.

I believe one of the Continental firms found it advisable to design a new series of engines, having a considerable longer proportionate stroke, thereby keeping up a fair piston speed at low r.p.m. as compared with land engines.

Regarding the weight of solid injection Diesel engines mentioned, I believe the weight of the submarine type works out about 55 lbs. per b.h.p., this type being designed with a view to lightness.

The Chairman: The last speaker's remarks on propeller speeds is a very important point. I think it is fairly well recognised by the makers of large engines, but perhaps not so much as it might be by makers of smaller engines. The revolutions of the larger size Diesel ships are comparatively low.

The Author's Reply: As regards the term "solid injection," I have used it simply because it is easier to say than "mechanical injection," which also is often used. Perhaps a better term still would be "airless injection." Still even that would not prevent people coining catchwords if it suited their purpose. The fuels used in the *Trefoil* varied almost from day to day, and often consisted of mixtures of all sorts of Admiralty boiler fuel, varying from heavy Texas to shale oils. We took fuel oil from tankers and from merchant liners, which carried fuel in D.B.s as cargo from all parts. I do not remember all names of fuel we took from these ships. We supplied the Fleet for boiler use. This same oil fuel we used for our Diesel engines and found it worked all right; no trouble. I hope, however, that the Institute will not be misled into believing that I recommend as an economical proposition the use of the more difficult fuels for marine engines. I think a lot of harm has been done in the past by claiming too much in this direction, and the seagoing engineer and the trade have suffered. There are oils in the hands of Oil Companies which cannot be burnt properly, even under boilers, and many directors who are not engineers will be only too glad to hand these over to engine makers who ask for trouble in the ships they fit out. A marine engine may have to run perhaps three weeks on end and then be liable to stoppages, starts, and reversals at the rate of one a minute for over six hours under the control of a canal pilot. This is very different from land practice. Putting together my experience with what I am told by other marine Diesel engineers, I am strongly of the opinion that overall economy is best served by avoiding residual oils with considerable incombustible constituents in marine engines. As certain parts of such fuel cannot be burnt, some use for these should be found. If an engine is used as a cross between a coking retort and a refuse destructor, it must be expected to behave as such. I know of a certain motor ship which has done regular running for some years now, in which it has been found that it is cheapest in the long run to use solar oil. I am not prepared to go quite as far as this, but taking into consideration the present attitude of some organisations regarding work on machinery in port, of which perhaps the less said the better, I would recommend a distillate whenever available. The fuel consumption of a Diesel ship is so small that she has a wide choice of bunkering stations. I may as well say, to disarm criticism, that the ship I refer to has port scavenging two-stroke engines with air injection, and at sea runs at about three-quarters of her rated power.

In reply to the question as to the form of the oil spray, perhaps a description of the spray in the open may give something towards the information sought. At the fuel pressures we use (about 4,000 lb.) the fuel comes out in almost uniform needles which will pass through a flame without igniting. At some distance from the nozzle these needles begin to look fluffy, and shortly afterwards rapidly expand into a cloud or mist which floats away. The jet can be ignited if a lamp be applied to it near the fluffy part, and a large sheet of flame like the tail of a comet takes place with a smacking noise. I suppose that when sprayed into compressed air the needle breaks up more quickly than in the open, but I believe an immediate breaking up would not give good results, perhaps because it is necessary to give a good speed to the flame as a whole with fairly high mean pressures. As regards reversing, the compression slows the engine very rapidly after fuel is shut off, and the engine is usually stopped by the time the reversing gear has gone over. The starting air applied for two or three seconds gets the engine running steadily in the opposite direction, after which a couple of cylinders are put on fuel and the rest is a matter of turning the wheel as rapidly as possible. All cylinders are fitted with air starting valves, and I think we should find this on most marine Diesel engines.

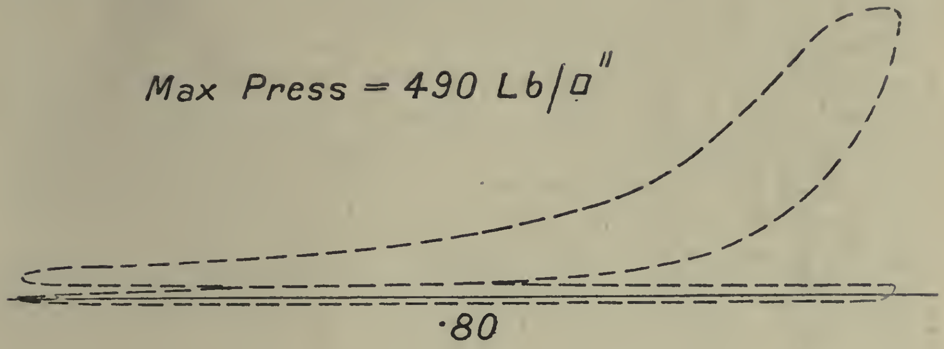
The bell crank bearings do not wear appreciably, as their duty is very low, and as a few degrees variation in the inlet and exhaust valve settings is immaterial, considerable wear would be necessary before refit is required. The dash pot I mentioned is an oil buffer to prevent too rapid a motion of the reversing gear, as suggested, I am not quite clear as to the question about hot pistons. The *Trefoil* engine is not a commercial engine, but a commercial engine of the same sized cylinder need not have water cooled pistons, and considerably larger engines are running with uncooled pistons. A piston is heated by the flame in the cylinder, and if not cooled internally, must pass most of its heat to the liner. If the cooling water to the liner fails through mud settling in the jacket or through other causes, the heat accumulates in the piston, both liner and piston heat, and the result is like that in an overheated bearing. I don't think there is any question of gases or acids.

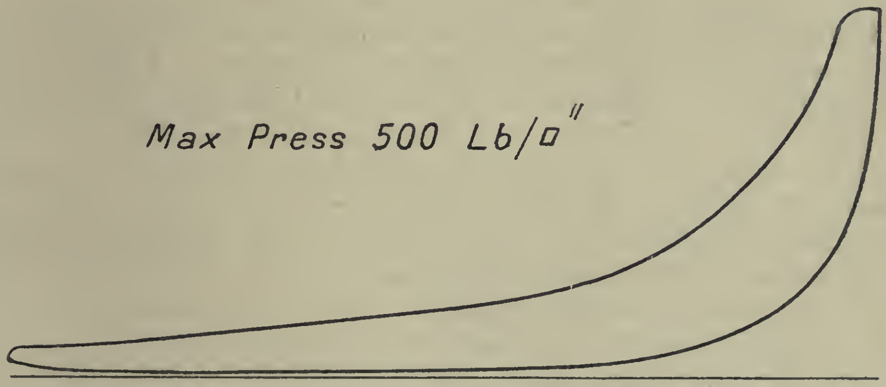
As regards the Diesel electric drive mentioned, if more than one engine is used for each motor shaft I suppose electrical complications may arise. If a single engine is used, the dynamo and motor seem to form a kind of electrical gear box, and the main advantage is that a smaller engine running at a higher speed than for the direct drive can be used. As long as the engine speed is moderate for its size this sounds quite promising, and I suppose first cost and durability would be the deciding factors. High speed oil engines are apt to be troublesome on long runs, however. Up to 150 r.p.m. or so, in a twin screw ship of ten or eleven knots, can, I believe, be used with propelling efficiencies equal to that of the ordinary single screw steam jobs. Our consumption I reckoned at about ·45 lb. per b.h.p.

I find I have no indicator cards, but I attach a tracing from one of the cards taken on shop test. The indicator ring was none too exact in my ship, so I have also added an actual overload card from the *Narragansett*, a twin screw 6-cylinder solid injection ship of 2,500 b.h.p., which may be of interest, and which I recently managed to get from that ship.

I think the member in his remarks on the number of cylinders is labouring under a misapprehension. The total b.h.p. of the *Trefoil* is 1,500, or about 94 per cylinder. As the cylinders are 17 in. diameter by 27 in. stroke, and run at 150 r.p.m., the mean brake pressure is 80·77 lb. This shows that the rating is higher than in most commercial marine engines. If the mechanical efficiency be taken at 78 per cent., which is probably not far wrong, as no compressors are fitted, the mean indicated pressure would be 103·5 lb.

With regard to the remarks by the Chairman as to the cost of upkeep and the time required for repairs in port, in the case of the *Trefoil* from the time she left the builders till she went for a refit, 18 months had expired, during all that time we were under one hour's notice to be ready for sea. I must say it was astonishing how these engines ran so long without having any repairs done—the only part of the engine we had adrift was one of the exhaust valve spindles, which had stuck; as for any other parts nothing had been touched, not even a bearing of any sort, in fact the main engines could have carried on another twelve months without a refit. Credit is thus due to the engine builders for their workmanship.





36 Hours Official Shop Trial

With reference to the question as to whether there was any corrosion in the air receivers due to gas or acid getting into those cylinders, I really could not tell you. All I noticed was that slight pitting took place through water being left, and that by careful attention to draining it off this action ceased. Whether there was acid or not we all know that there is a tendency for pitting to take place where water lies in a partly filled boiler, and that in these receivers, as water is carried over with the air we have to pay particular attention that they are carefully drained and watched.

THE SOLID INJECTION ENGINE.

The following articles are quoted from the *Motorship* for September, 1919:—

EDITORIAL.

The question of solid injection.—Pioneers in modern engineering make advances so rapidly that the other manufacturers and users are occasionally confronted with situations that to them are problematical from a business point of view. The result is that the question of meeting these engineering advances is sometimes shirked as long as possible. This has been particularly so in the case of the heavy-oil engine and its construction and use aboard ship, and some vessel owners and shipbuilders are still endeavouring to avoid it. They may hold out until its certain and extensive adoption by others, threatens to jeopardise their trade. There is a limit to the time before them; as in this age of continuous progress, sound engineering developments which are practical commercial possibilities, will never be ignored for long.

In a similar way, a problem now confronts all builders of marine Diesel engines. This is the question of the solid-injection of fuel for high-compression oil-engines. Until recently it was the general concensus of opinion among oil-engine designers that this system was uneconomical and of little use for the heavier grades of fuel-oil. In fact, up to the commencement of the war this was known to be the case. But during the last few years important progress has been made abroad which demands closer investigation and experimenting on the part of our own engine builders. It has been thought that only one English concern was using solid-injection so it may be well to mention that for several years as many as twenty other engineering companies in Great Britain have been using this system under licenses. This, by the way, is the largest number of constructional licenses granted for any one make of Diesel engines. It indicates the technical opinion of these engineers familiar with the system.

According to our information a consumption of 0·28 lbs. per indicated horse-power (0·38 lbs. per b.h.p.) can regularly be maintained and an even lower consumption has been obtained on the test-bed. This, of course, is remarkable. The economy resulting from such achievements may be compared with the best marine steam-turbine practice; namely, 0·95 lbs. per indicated horse-power, or with 0·30 lbs. for the best air injection Diesel practice.

With four 6,000 i.h.p. fast cargo ships of the same power fitted with coal-burning reciprocating steam-engines, oil-fired geared turbines, air-injection Diesel engines and the solid-injection Diesel engines respectively, we get the following daily fuel consumptions:—

Coal-Burning Ship	$96\frac{1}{2}$ tons
Oil-Fired Turbine Ship	61 ,,
Air-Injection Diesel Ship	19 ,,
Solid-Injection Diesel Ship	18 ,,

It will be seen from these figures that solid-injection widens the gulf between oil-fired, steam and internal-combustion engines, but that the actual difference in consumption between it and the air injection system is not very much. Of course, it dispenses with the air-compressor and saves the power absorbed in that manner, and, on the basis of indicated horse-power there is a gain of about 5·2 per cent. efficiency per ton of fuel consumed over that attained in engines using air injection.

The figures would have been more effective had we quoted the consumptions per brake horse-power in each instance. Usually steamship owners carry in mind the consumption per indicated horse-power; so it may be best to use the more familiar term.

Perhaps it is well to make it clear that there is no desire on our part to suggest that builders abandon air-injection in favour of solid-injection at this stage. We simply intend to indicate the necessity for immediate investigation and experiment. Our own view is quite impartial, except that we naturally lean towards air-injection because we are acquainted with both its merits and weaknesses, whereas we do not yet know the faults of solid-injection. While abroad we hope to have the opportunity of making further investigations.

PRACTICAL OPERATION OF MARINE DIESEL ENGINES.

By K. Carlsen.

The popular objection to the Diesel engines for marine propulsion is that the men necessary for their reliable operation are of a high grade, and consequently, are difficult to obtain. It is acknowledged to be poor policy to place a steam engineer in charge of the machinery of a motorship without a preliminary shop course and some erecting and testing experience. To cover the ground briefly, the difficulty does not arise from the complications in the machinery, but from the necessity to have every detail of each working cylinder in proper adjustment. True, the adjustments must be finer and more exact than for steam, but there is considerable simplicity obtained by the fact that the pistons, rings, valves, valve gear and cams are all alike for each cylinder.

It is exceedingly simple to operate a Diesel engine when it is in correct adjustment, and almost every engineer prefers the Diesel engine to steam machinery after being able to make comparisons from personal experiences.

The reliability and cost of maintenance of the Diesel engine depend, as with steam machinery, on the care and "engineering sense" of the personnel. The engines should be kept in the best possible working condition and any defect remedied immediately.

[Before going further it may be remarked that the problems of ordinary engine-room routine work whether on steam or Diesel engines are really those of diagnosis of symptoms which may come to the attention of the watch officer through his five senses. Troubles are nearly always noticed either from the feel or the sound of some part working improperly. Sometimes that so-called sixth sense will detect trouble before it really arrives.—Editor.]

STARTING.

If any repairs or adjustments have been made previous to the time of starting, the engine should be turned over by hand or by the turning engine. All moving parts should be clear of everything and all valve timing and manœuvring gear should be checked for correct operation. All lock nuts and split joints should be drawn up tight. Valve spindles should all work easily, but without leaks. The lubricating oil tanks and fuel oil settling tanks should be measured or otherwise tested. The condition of these tanks must be known, for clean lubricating oil and fuel is essential to reliable operation. The operator should give the inlet and exhaust valve stems a little kerosene. He should be sure that all overflow valves in the fuel line and also the indicator cocks are closed. To test the fuel line and to insure reliable fuel injection after starting on air, proceed as follows: Test the fuel line for leaks by opening it up to the high pressure air in the spray air bottles. Then close the fuel line to the injection air. Open the drain valves to let out all air pressure and immediately close them again. At least 800 lbs. pressure should show on the gauges for the spray air bottles and a good supply of starting air should be available.

Different designers use different pressure and amounts of starting air, but few engines of the full Diesel high-compression type will start when cold on much less than 150 lbs. of air.

All lubricating devices should be filled with oil and regulated for proper feeding. The oil pumps for forced lubrication should be started if separate from the main engine. If necessary, the cranks should be placed in the proper position for a positive start. The inlet valve, if any, on the low pressure side of the air compressor, should be opened wide, together with all drains from the inter-coolers.

[Evidently the writer means that the throttle valve in the first-stage suction of the air-compressor should be wide open—Editor.]

If the temperature of the cooling water is very low, do not let it run through the cylinder in a full stream or for any length of time before starting. The cylinder might be cooled to such an extent that it will be difficult, if not impossible, to start the engine. All valves on the fuel supply line to the fuel pump should be opened wide. Adjust the fuel pumps to work by hand and pump oil into all fuel lines till it appears in a solid stream from the overflow valves. Then stop pumping, close all overflow valves and put the hand pumping gear out of action. Open the valve between the H.P. compressor and the air bottle and the engine is ready to start.

After the engine is started, open the cooling water discharge from the circulating pump to the running position and close all the drains from the inter-coolers. Fill all air tanks to the right pressure as soon as possible and examine the engine all over to be sure everything is working properly. [Of course, anything that has been adjusted since the last run will be carefully watched for a time—Editor]. If the pistons are water-cooled, inspect this system and ascertain the volume and temperature of the cooling water discharge. All cooling or circulating systems should be started before or at the same time as the engine. Everything proving all right, the engine may be stopped and it will be in readiness for immediate service.

If compressed air is not available, compressed carbonic acid may be used to fill up the bottles, but under no circumstances may oxygen, hydrogen or any other gases be used. In filling a bottle with carbonic acid gas all valves should be closed, while the carbonic acid tank is connected to the air bottles. Open the valve between the two tanks. The pressure in a carbonic acid tank is usually about 750 lbs. per sq. in. and when the pressure has dropped to about 300 lbs. close the valve and slowly pour hot water on rags laying around the top of the carbonic acid tank. When the pressure has returned to about 750 lbs. open the valve to the air tank again. Repeat this until all the acid is evaporated or until the required pressure in the tank is obtained. After the engine is started, immediately fill all the tanks with air and drain out the rest of the carbonic acid. If the injection air tank has been filled in this way, the exhaust valves should be cleaned as soon as possible.

WHILE RUNNING.

The most important things to watch while running, are the different gauges. These will often show where to look for trouble if there is anything wrong. The engine should run evenly without smoking or knocking. Care should be taken to see that the cooling water circulates properly through all parts requiring it. Small cocks are generally fitted in the cylinder heads to let air out of the cooling spaces and thus avoid air pockets which prevent proper cooling. Thermometers are furnished at different places where a constant check on the temperature is necessary. The temperature of the cooling water at the outlet should not be so hot that it cannot be felt by hand. 160° Fahr. is a maximum. The lubricating and fuel oil filters should be cleaned at frequent and regular intervals. The different stages of the air-compressor should each do an equal part of the work in compressing to the highest pressure. If the pressure in the forced lubrication system drops, it is usually on account of dirt in the filter. The daily supply tank should be drained frequently because excess water in the fuel oil will stop the engine. There are generally two supply tanks. As soon as one is empty, the other may be connected to the pump suctions and the first filled up immediately. Any water in the fuel will thus have time to settle down, and be drained off. Air bottles should be drained of condensed water and there should be two drain valves; the one next to the bottle should be fully opened and the other just cranked open a little to let the water out.

Draining water through a valve opened very slightly and with two valves in the drain pipes, the first may be closed tight and the second taken off and ground when necessary.

Always be sure that the water or oil circulates properly if the pistons are cooled. If the oil circulation is not sufficient, the oil will carbonise, resulting in the piston heating and the rings sticking. Press fingers on all valve rollers and if the clearances are correct, they can be prevented from revolving for the period during which the cam is not under the roller. [In some designs of valve gear the valve roller is intended to roll on the cam at all times.—Editor].

After the engine is stopped, close all valves on the fuel supply line. The cooling water should be gradually reduced and not stopped until the engine has cooled down sufficiently to prevent the lubricating oil from drying on the cylinder walls. In a cold climate, if there is danger of the water freezing, all jackets and pipes should be drained. All lubricators should be turned off and the force feed lubricating pump stopped. All valves to the air injection and starting tanks should be closed tight, and the air blown out from the injection line and compressor. None of the gauges should show any pressure.

Will Not Start.

If, when starting the engine, combustion does not take place it cannot be forced by prolonged operation on air. The only result will be loss of valuable starting air. The difficulty is generally due to one of the following reasons:—

I. No fuel.

(*a*) The fuel valve from the daily supply tank may be closed.

(*b*) Daily service tank may be empty

(*c*) There may be air pockets in the fuel supply line or in the fuel pump. This might happen if the supply tanks were not high enough above the suction valve of the fuel pump. In this case, take off plug above the pump's delivery valve, after closing the valve to the supply tanks, then open this slowly and let the oil run out until no air bubbles escape from the outlet. Then fill up the line to the fuel oil valves in the usual way.

(*d*) Fuel may be leaking past the fuel pump pistons.

(*e*) The line between the pump and the fuel valves may be empty, either on account of improper filling of the line, or from leakage if too long a time has elapsed before starting after the line is filled.

(*f*) The hand pump may be loose.

(*g*) Sometimes the engine will not start if the atmosphere is at a very low temperature. It may then be necessary to fill the water jacket with steam or heated water.

(*h*) At such low temperature the fuel oil may be too thick to run. Therefore, ships that have to go to cold climates should have arrangements provided for heating the fuel oil.

II. Low compression. The temperature may then be too low for ignition.

(*a*) The slots in the inlet pipes may be filled with dirt.

(*b*) Compression may leak past the valve housing and valve seats.

(*c*) The striking joint may be too big.

The various troubles that may develop in the running of the engine may be classified by their symptoms. In fact, the same result may be caused by totally different defects and it is the test of a good engineer to find the true cause promptly.

I. Motor knocks while running.

(*a*) A loose bearing. The knock may cease if the bearing is flooded with heavy oil.

(*b*) Too early injection. This may cause slight explosions instead of proper combustion and is to be remedied by the proper adjustment of the fuel gear.

(*c*) Too late injection.

(*d*) Too high or too low injection air pressure.

(*e*) The fuel valve may be leaking badly, and in that case the engine should be stopped immediately. Under this circumstance the fuel is getting in the cylinders too early and there may be ignition before the piston is at its top centre.

(*f*) Improper pulverising on account of dirt in pulveriser. Raising the pressure of injection air for a short time may remove the dirt.

(*g*) Improper distribution of fuel to the different cylinders.

II. If the engine is working correctly, the exhaust gases will be nearly invisible. If the exhaust gases are smoky or black, this will be due to following causes :—

(*a*) Too low injection air pressure.

(*b*) The fuel valve is not lifting high enough. This can be remedied by raising the injection air pressure. [Or reducing the clearances in the valve gear.—Editor].

(*c*) Leaking fuel valve. The valve may stick or need grinding. It may be on account of improper adjustment, which would prevent the valve from closing properly. In this case it is difficult to keep the injection pressure, because the air is continually leaking into the cylinder. This should be remedied immediately.

(*d*) If overloaded, the engine will smoke. This is not necessarily due to faulty operating. If the overloading of the engine will be necessary for some time, raising the pressure of the injection air, or a slightly earlier injection will lessen this trouble.

(*e*) The pulveriser may be dirty.

(*f*) A hot bearing will overload the engine and make it smoke; careful investigation may locate this trouble and increased lubrication overcome it.

(*g*) Resistance against the inlet air. The slots in the inlet pipes may be partly closed with dirt and must be cleaned.

(*h*) Resistance against the exhaust gases. The exhaust pipe is dirty or there is water in it.

(*i*) The compression is too low.

III. Uneven Running.

(*a*) The governor connections are not working freely; it should be re-adjusted.

(*b*) The air compressor pistons may be leaking, or the automatic valves are leaking, sticking, or not seating properly. In this case, the air injection pressure might vary.

(*c*) The fuel pump is working irregularly on account of dirt in valves or leaky stuffing boxes, or it may be working too fast on account of improper adjustment.

(*d*) Sometimes each cylinder is not doing an equal share of work. This will be easily found by taking indicator cards, and is usually on account of improper distribution of the fuel oil.

(*e*) If, when opening the test holes in the exhaust pipe, the exhaust is not uniform in colour and sound, the improperly working cylinder should be examined with regard to valves and their adjustment corrected.

(*e*) Water in the fuel will make uneven running, if it does not stop the engine.

IV. If the air pressure cannot be maintained, the engine is using too much or the compressor is not delivering enough.

(*a*) Count the r.p.m. for if the engine is running slowly it uses more air because the fuel valve is open a longer time.

(*b*) If the engine is running at the proper speed it might be using too much air because the play between the cam and roller is too small. The lifting of the valve would then be too great.

(*c*) The fuel valve may stick or lead. Stop immediately and fix it.

The compressor may not be delivering enough air for following reasons:—

(*d*) The intake pipe, if any, on the compressor's low pressure side, is plugged up with dirt, or the valve is not fully open.

(*e*) The valves are leaking or sticking. This can be determined by laying the ear on the compressor close to the valve cages.

(*f*) The striking joint may be too big.

(*g*) Leaky or sticking piston rings.

(*h*) Leakage of air to cooling water.

(*i*) Leakage of air at the forced feed lubricator.

(*j*) Leaks in pipes, valves or connections. These last can be distinguished by a whistling sound.

If there is oil in the cylinder before starting, it should be taken out. There can of course be no more oil burned than that which will use up the air in the cylinder and for that reason the pressure resulting from an overcharge of oil will hardly ever exceed 1500 lbs. per sq. in. This is usually a safe pressure for the working cylinders.

However, the ignition and consequent combustion may occur early in the compression stroke and further compression result in excessive pressures. Usually this results in blowing through the cylinder head gasket or down past the rings. This is only likely to occur on starting when the accelerating forces are high, due to the starting air pressures and the extra heavy charges of fuel from the primed valves. Always close the valves in the fuel line to the injection valves upon stopping for a reasonable length of time.

The above seems to be just a long list of errors and rules, but on looking them over it will be seen that they embrace nearly the same remedies over and over again. These remedies are very simple and quickly applied, the principal source of trouble being the timing, adjustments, and tightness of the valves. The proper working of the fuel valve is most important.

SESSION 1921-22.

President: SIR JOSEPH P. MACLAY, BART.

Some Observations on Marine Oil Engines.

BY MR. D. M. SHANNON.

Read Tuesday, March 15, 1921.

IN looking back over the progress made by the heat engine during the last century, two notable failures can be recorded against the engineer. The first is that no heat engine has been produced which can utilise small drops in temperature at or near atmospheric temperature, and the second is our inability to evolve a cycle that will give a higher efficiency than that expressed by the well-known equation: $\frac{T_1 - T_2}{T_1}$

A practical solution to the first problem would enable us to use large stores of heat that are now lost, and incidentally might open up the prospect of being able to propel a ship by means of the temperature difference between the deck and the keel. The benefits of a decided improvement in the thermal efficiency of the heat engine are best left to the imagination.

Our present conception of the heat engine is bounded by the rudimentary idea that the application of heat to a body causes it to expand and gives it power to do external work when properly harnessed, and until some other method is found of converting heat into potential, it is unlikely that much improvement can be hoped for. The state of our chemical knowledge does not give much encouragement that we will be able to devise a cycle working on endo-exothermic transformations, and the introduction of the turbine principle will at most, only give us the same relative advance in economy and simplicity which it brought about in the case of the steam engine.

The chart given on Fig. 1 shows the improvement made in the thermodynamic efficiency of the heat motor during the last century, and presents many of the great inventions in their correct perspective. The curves indicate the main lines of progress of the commercial heat engine and ignore some results which, although very promising, failed to establish themselves for reasons other than thermodynamic ones.

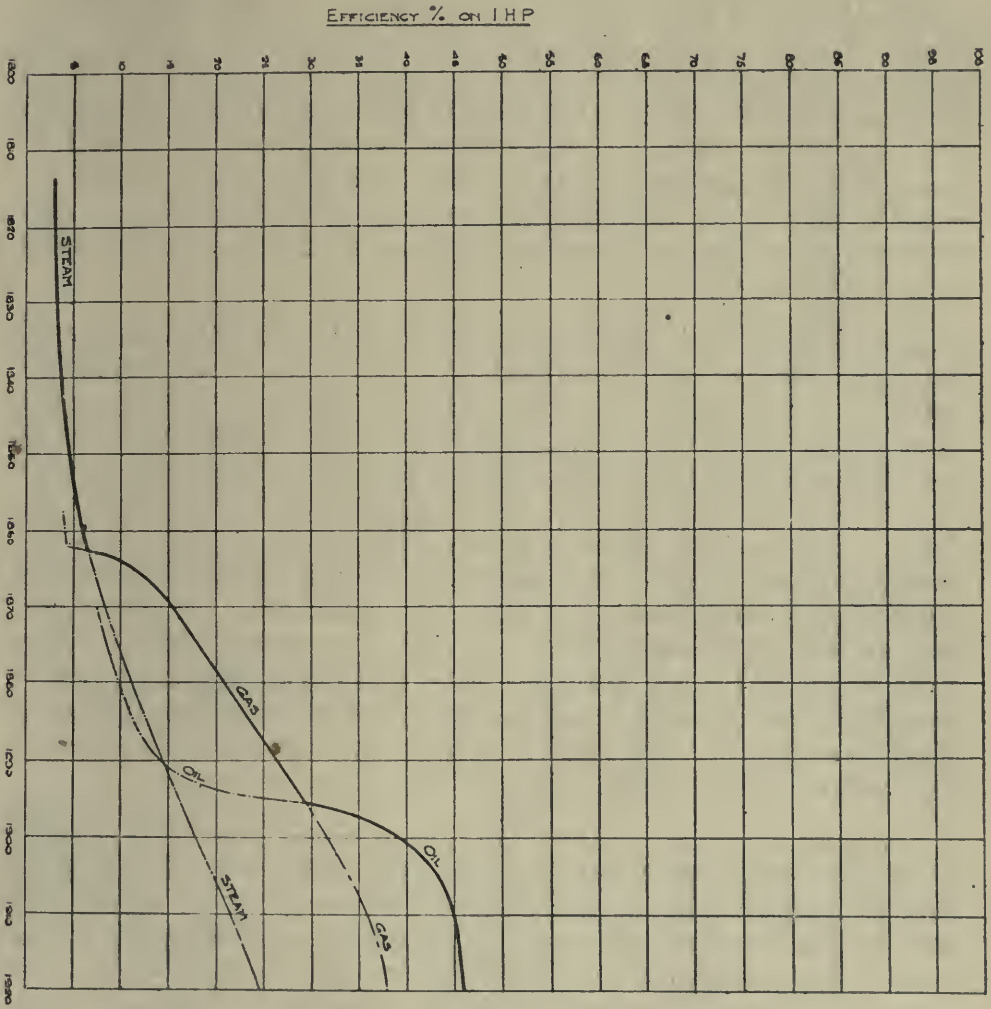


Fig. 1.

Watt more than doubled the efficiency of the steam engine, and since his day no invention has brought about any relatively great improvement in its performance.

When first made a marketable machine by Lenoir, the internal combustion engine in the form of the gas engine, was a retrograde step, but the introduction by Otto of compression before ignition, first proposed by William Barnett in 1838, sent it ahead of all competitors. To this Englishman belongs the credit of having brought out the greatest heat engine invention since the days of Watt.

Similarly, the oil engine started behind both the gas and the steam engine, and it was not until the coming of the high compression self-ignition engine, which although not invented by Diesel was brought to its present position principally by his labours, that it forged ahead of the gas engine. It should be noted that the gas engine curve does not take into account the considerable losses in the producer, and that the 100 per cent. line is put on the diagram just to show that we are still a long way from the top.

Also, no account is taken of the possible increase of say 10 per cent. which can be obtained with the gas and oil engine by using suitable jacket and exhaust heat recovery methods, the possibilities of which have as yet only been touched upon. The heat losses are so simple and definite that it is surprising greater success has not attended the efforts that have been made to tap them. Not so the steam engine. It has made use of these devices to the full, and it is solely by the aid of such tonics and revivers as economisers, super-heaters, feed heaters, condensers, etc., that it has been robbed of a decent funeral. If the ingenuity spent in trying to keep it alive had been spent on improving the more promising motor and in following up Barnett's ideas, our curve of efficiency would have been much nearer the top to-day.

The progress of the heat motor from the embryo of Hero to the adolescent stages of Watt and Stephenson was one of comparative simplicity when one considers the amount of effort and money spent on its later stages of development to bring it to the present day comparatively high efficiency internal combustion engine. Many types have come and gone, and more and more improved forms have successively emerged until one wonders why the steam engine is enjoying such a healthy old age. Engineers appear very reluctant to exchange their trusty old friend for one that is after all, much more fickle and difficult to understand.

Failures have been recorded, but unfortunately most have been hidden, so that the lessons to be learned from them have been lost to the cause. Had these been carefully charted for the use of others, progress would have been much faster and less costly. It is after all much better to be guided by lighthouses than by the vision of wrecks.

Probably the fact that coal has been too cheap in this country has blurred our vision and stunted our efforts, and the incentive to study its most economical use has therefore not been sufficient. We could be truthfully classed as the greatest coal-wasting nation that has ever existed.

A ton of steel when built into a ship is worth about £22, and when made into hair springs for watches is worth about £6,000,000, from which it would appear that it is only because there is an over-production of steel for the watch- making trades that we are able to build ships and engines. With coal, however, the matter is different. It costs about £2 per ton to buy, yet it contains derivatives, certain of which are valued at £24,000 per ton. These are destroyed in open fires instead of being recovered by low temperature distillation, and we are presented with the spectacle of a poverty stricken nation warming itself over a fire of bank notes.

The marine engineer has some difficulty in deciding as to the comparative advantages of the numerous types of oil engine now made. Our views are apt to be coloured by our interests, and it is a rare gift to be an active combatant and an impartial umpire at the same time. The tendency of to-day is that merit is being replaced by monopoly and many a new idea is fought and killed because it might adversely affect some financial, or other interest.

Whilst the four-cycle engine is in the strongest position at the present time, it must not be taken as evidence that it is the best. The law of evolution makes it compulsory that every improved invention must start off in the minority. One cannot afford to be dogmatic when speaking of human progress, and although a certain thing persists to-day a new idea may already be forming somewhere, which will cause progressive men to change their minds, finality in this as in every other art, being as far off as the end of the earth.

In this subject simplicity will count for more than past performance, and if the four-cycle engine is to maintain a leading place in the future it must be made double-acting. Whilst this would reduce the number of cranks and connecting rods and improve its position as regards weight and space, it would still lack the delightful simplicity of its almost valveless contemporary.

This valvelessness of the two-cycle engine is a feature that will have a large say in bringing about the wider application of the newer motor and is a point more appreciated by the running engineer than by the owner. For instance, the engine in the motor-ship *Fullagar,* which has a total of four working valves as against 18 required for a similar sized four-cycle engine, did not require any valve touching in any way during six months running.

Amongst British marine engineers the tendency is to adopt the two-cycle system, about six firms being interested in the four-cycle engine whereas about thirteen have gone in for some form of two-cycle motor.

This is an important indication of the way the current is running. It is also certain that more will be heard of the double-acting two-cycle engine similar to, but with considerable improvements over the type fitted on board the ex-German motorship *Fritz.*

The Diesel and the hot-bulb engine practically monopolise the whole of the merchant marine application of the oil engine, the latter, if anything, gaining ground at the expense of the former. This is simply due to its low initial cost which is bought at the expense of sacrificing features which ought to be strictly observed in marine practice. The hot-bulb engine has made practically no progress in design during the last ten years, and much valuable experience obtained with the two-cycle Diesel engine has been entirely ignored. The exhausting and scavenging of such engine is not all that could be desired, there being ample room for considerable improvement in the arrangement of the inlet and exhaust ports and in the distribution of the time areas through them.

In order to improve the output and render the main working parts open and accessible so that ordinary drip, or forced lubrication of a copious nature can be fitted, the simple but objectionable and inefficient feature of crank chamber compression will have to give place to a proper receiver scavenging system, in all but the smallest engines.

SOME OBSERVATIONS ON MARINE OIL ENGINES.

Fig. 2 shows diagrams taken from the crank chamber of a hot-bulb engine. The length " B " represents the engine stroke, whilst " A " represents the fraction of air passed through the cylinder. At a speed of 240 r.p.m., 85% of the air compressed in the crank chamber was passed through the cylinder, and at 510 r.p.m. this amount had fallen to 37%, showing the hopelessness of this system for high speed work.

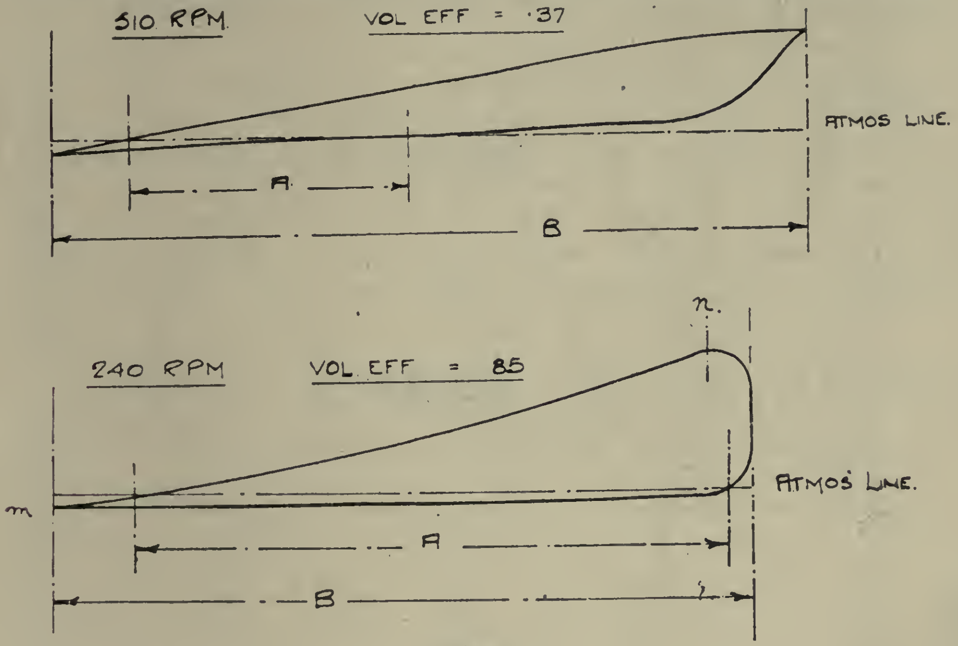


Fig. 2.

Fig. 3 shows a diagram taken from a high efficiency scavenging pump, such as is fitted to the opposed piston engine of the *Camellaird-Fullagar* type, which should be compared with the lower diagram of Fig. 2. This latter indicates a steady rise in pressure from m to n at which point the air ports are uncovered by the piston, and the pressure falls rapidly to zero. The former shows a rise from o to p at which point the receiver pressure is reached, the discharge then remaining approximately steady for the rest of the stroke, *i.e.*, the inefficient top of the lower diagram of Fig. 2 has been cut off by the receiver effect. The

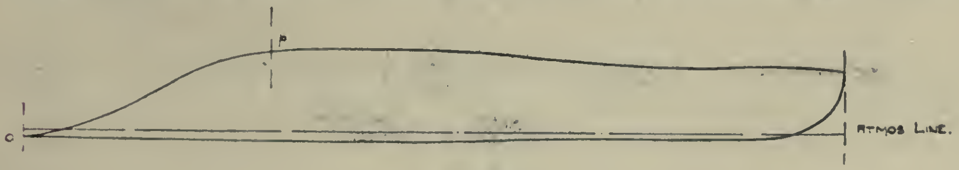


Fig. 3.

obvious way of improving, say the ordinary 4-cylinder hot-bulb engine is to fit piston rods and use the under side of the pistons as ordinary scavenging pumps having inlet and discharge valves, and all discharging into a common receiver connected to the inlet ports of the cylinders. In this way the supply to, and the take away from the receiver is even, and scavenging of the cylinders would be carried out at a low and fairly uniform pressure, which are the best and most efficient conditions.

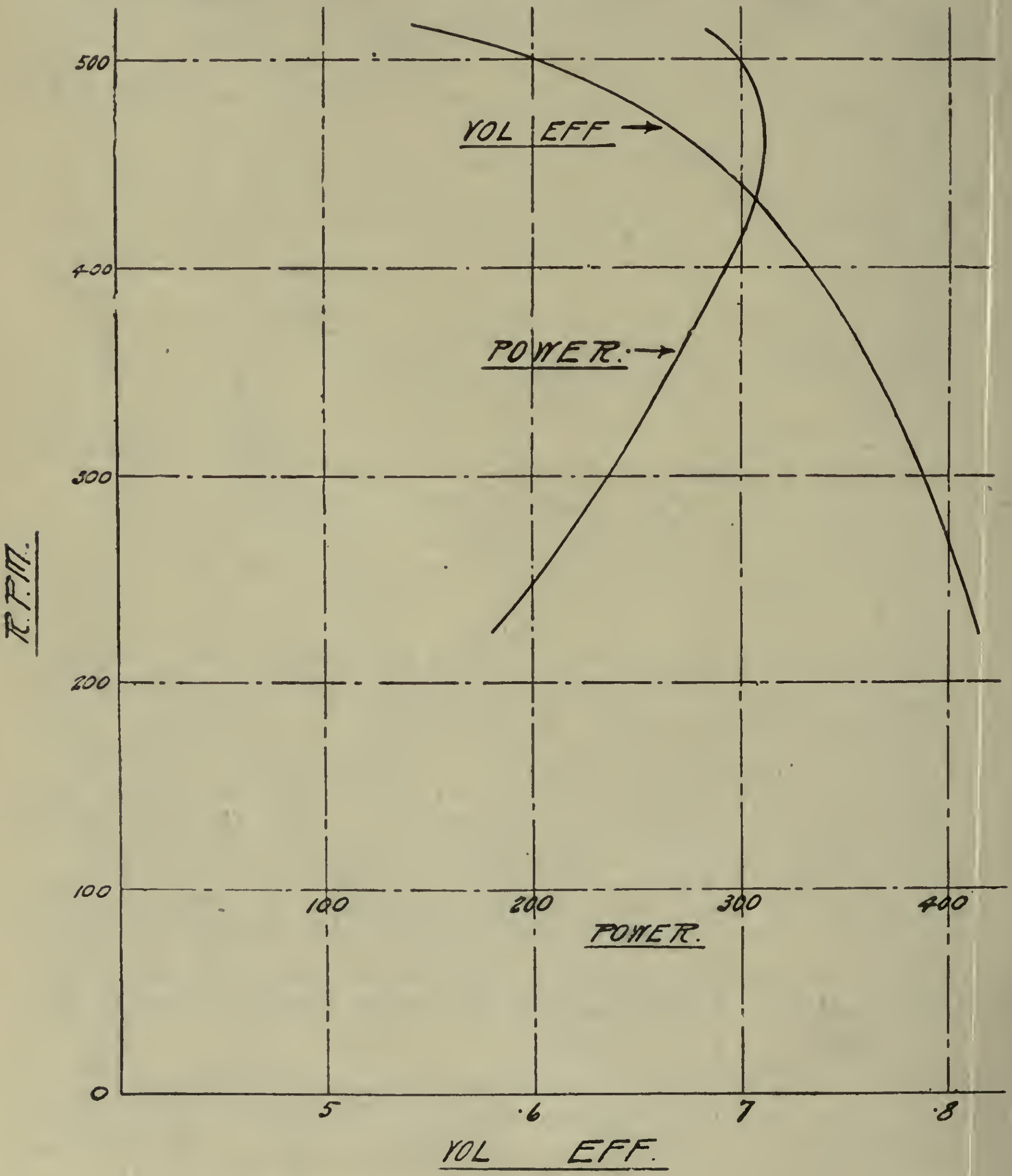


Fig. 4.

Fig. 4 shows the curves of volumetric efficiency and power plotted from tests on another engine, and further illustrate the inefficiency of the system, the power actually falling off above a certain speed.

There is a tendency at present amongst designers to increase the number of systems on which it is possible to make the internal combustion engine work. Cycles are becoming embarrassingly numerous, the dividing line separating one type from the next on the series being so narrow, that fractionating becomes difficult. Having regard to the nature of the materials with which we are working, it should not be expected that any engine will work on what could honestly be called a true cycle, and any attempt at classification can be looked upon more as one of convenience than of scientific accuracy, the distinction being useful to our limited knowledge. For instance, a change in the weather may put one that is advertised as a cold starting engine into the category of the preliminary heated ones.

Diagrams from various types of piston engines are illustrated by Figs. 5 to 12. Fig. 5 shows a diagram taken from a four-cycle submarine petrol engine working on the Otto cycle.

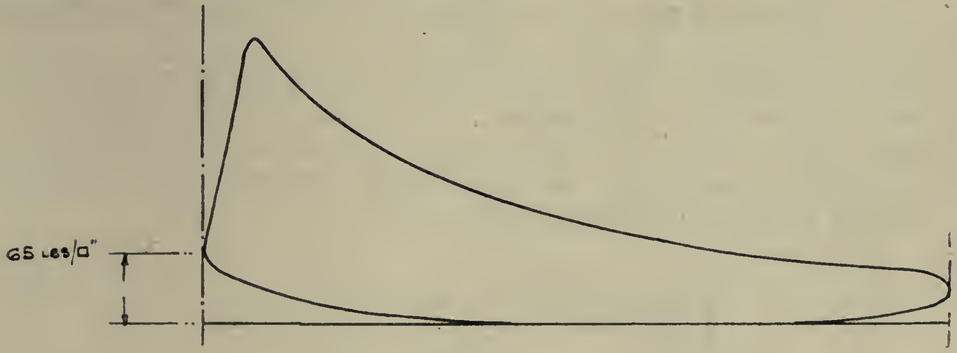


Fig. 5.

When the hot-bulb engine was evolved, an endeavour was made by preliminary heating a part of the combustion chamber and using a compression pressure of about 150 lbs. to get constant pressure combustion at that pressure, and so reduce the loads on the working parts of the engine in order to give a light and cheap machine.

With an air injection hot-bulb engine the writer was able to get the card shown by Fig. 6, but unfortunately the high fuel consumption did not permit of putting an engine on the

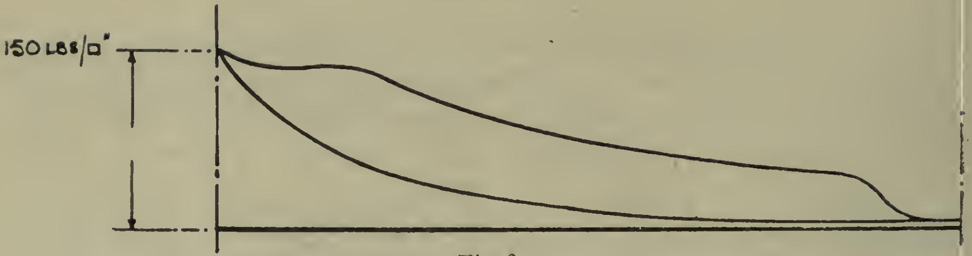


Fig. 6.

market working on this cycle. Fig. 7 shows the best form of card obtained from this and all other hot-bulb engines when working at best efficiency, the sudden rise in pressure when the

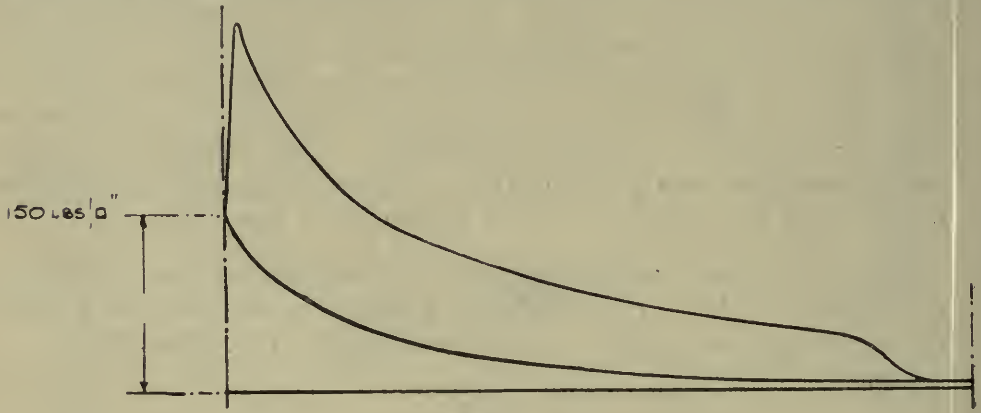


Fig. 7.

fuel is admitted being necessary. Fig. 8 shows another card from this engine. The mean pressure of 77 lbs. per sq. inch is high for a hot-bulb crank chamber compression engine giving smokeless combustion, and cannot be equalled by any similar engine working without air injection.

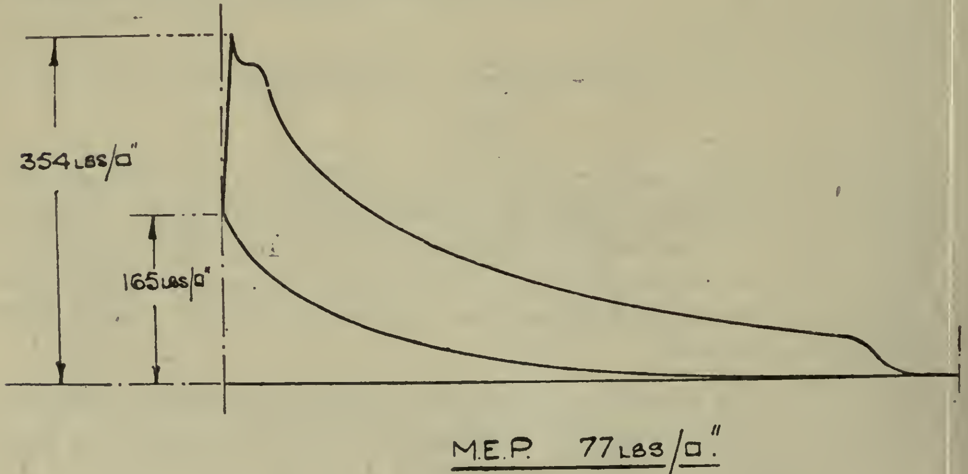


Fig. 8.

Fig. 9 shows a card taken from a Diesel engine working with mechanical injection on Scotch shale oil, and shows very distinctly the dual cycle, *i.e.*, part combustion at approximately constant volume and the remainder at approximately constant pressure. Such cards as a rule are the result of accident rather than design, and cannot be realised economically in engines having a single injection system with existing

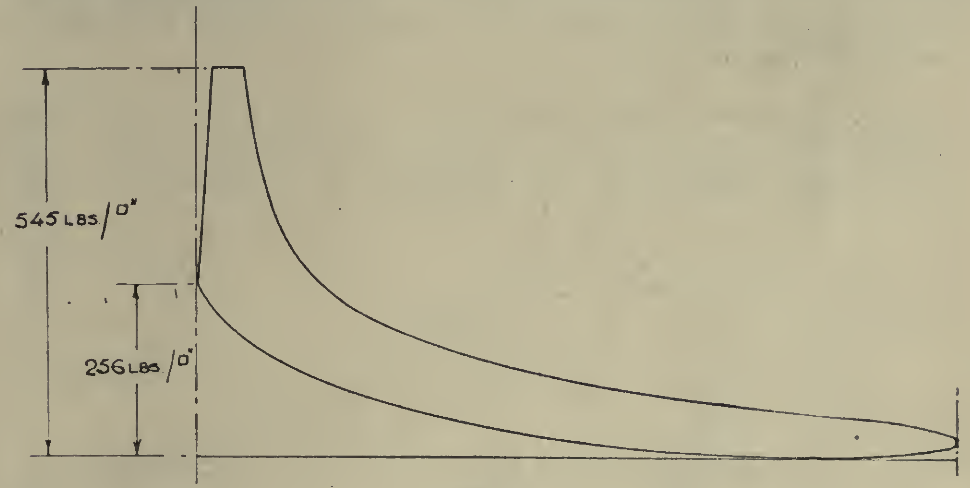


Fig. 9.

designs. The maximum pressure has to be increased to about 600 lbs. and the form of diagram as shown by Fig. 10 obtained,

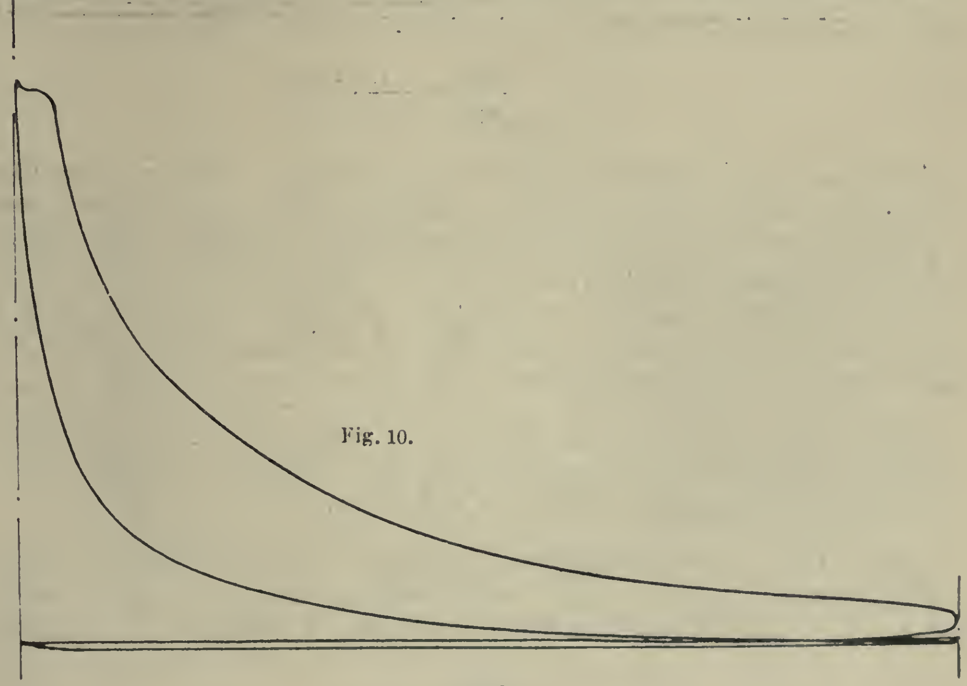


Fig. 10.

before the fuel consumption is such as to permit a marketable engine.

Any advantages claimed because of the lower compression pressure required by mechanical injection Diesel engines are largely fictitious, as the maximum pressure in the cylinder nearly always exceeds that given by air injection, and this is the pressure that matters when the safety of the structure is to be determined.

Fig. 11 shows a card taken from an engine which was constructed about 10 years ago, working on the Brayton cycle. All the air for combustion was supplied at a pressure of about 300 lbs. by separately driven compressors. Some of the exhaust gases from the previous working stroke were trapped and compressed to about 250 lbs. per sq. in. to obtain the temperature necessary for combustion. The engine was of the two-cycle single-acting type and gave perfect combustion and a good fuel consumption, but failed to emerge commercially on account of complicated design and infantile weaknesses.

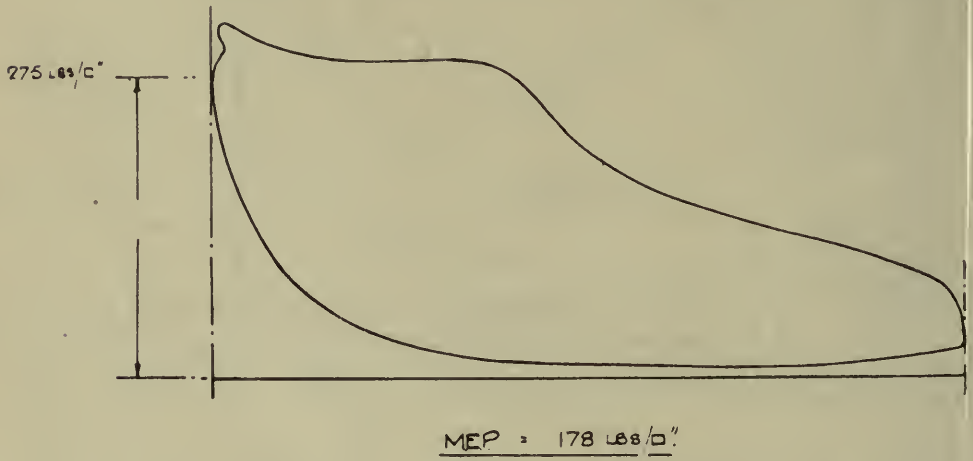


Fig. 11.

The mean pressure of 178 lbs. is noteworthy and has only been exceeded by the opposed piston engine working with super-scavenging and with two fuel valves per cylinder.

Fig. 12 is an overload card taken from a *Camellaird-Fullagar* engine with one fuel valve, and working without super-charging. This indicates that there is a large margin of power with this type of engine, of which advantage may be taken in many ways.

It will be noticed that the method of injecting the oil plays a considerable part in the classification of the oil engine and, that so far, no large motor has been produced with a system of injection which would allow it to work on low compression with spark ignition.

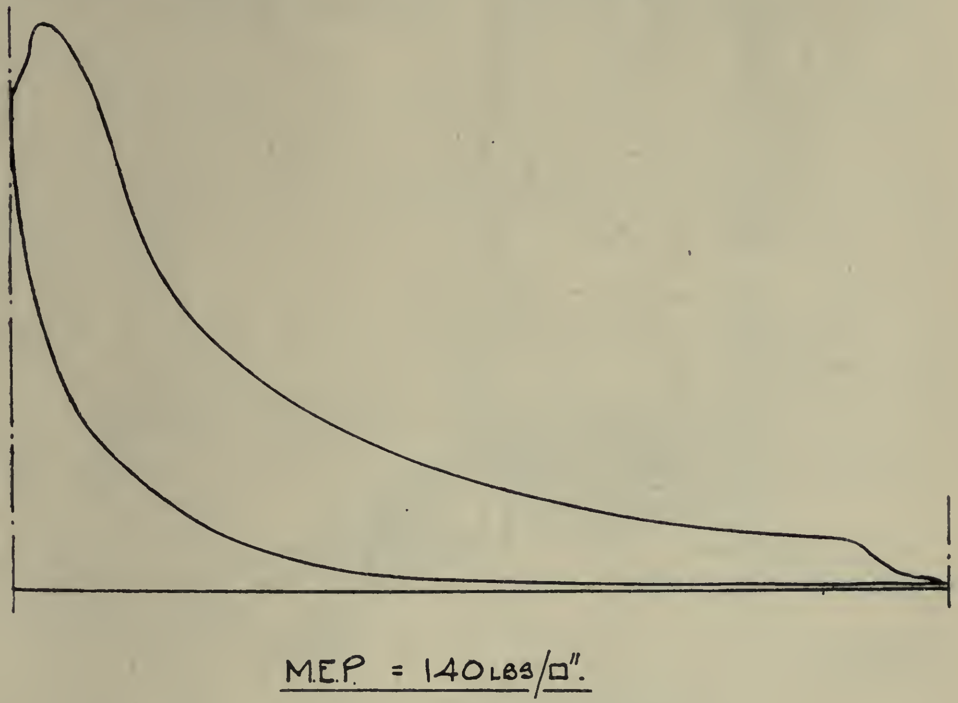


Fig. 12.

In the order of importance the air ignition system is an easy first, this method practically dominating the entire field of Diesel engine application. The air compressor has had a good deal of unjustifiable adverse criticism which had its foundations in a wrong conception of compressor design and duties, and carefully ignored the success obtained by many firms. Its performance on the hundreds of motor ships and land engines has been wonderfully satisfactory, and no case can be made out against it on the score of unreliability. Whilst exhaust valves, where fitted, and fuel valves of all Diesel engines, require frequent attention, the compressor has only to be examined at very long intervals and is by far the part of the engine which costs least for upkeep.

The resuscitation of mechanical injection owes much to the very bad performance of an early compressor, the principal dimensions and peculiar features of design of which are shown by Fig. 13. There are no inlet valves on either stage, the air

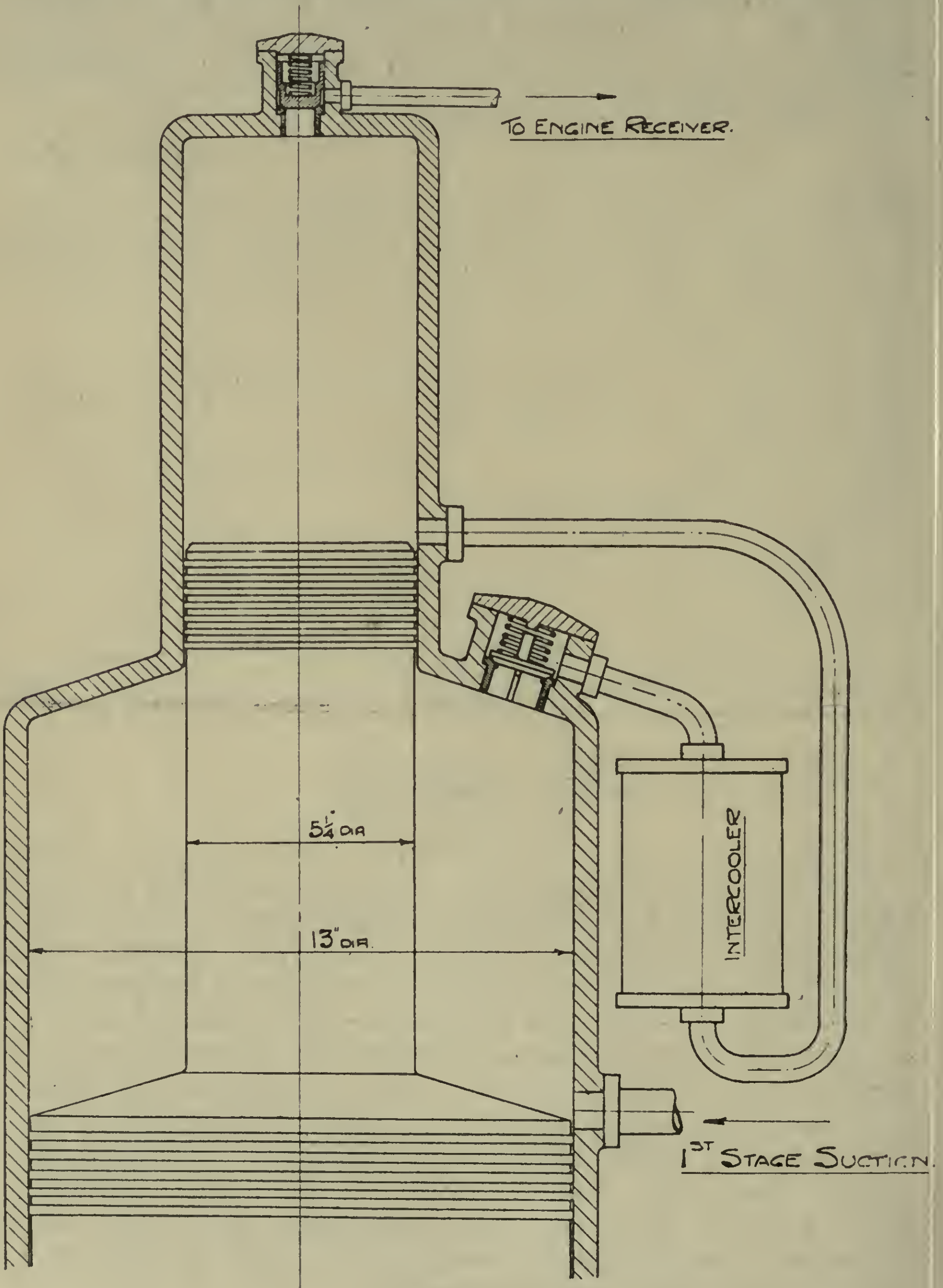


Fig. 13.

being admitted by ports uncovered by the pistons. The loss due to this system was about 27 H.P. or about 4½% of the engine power, the fuel consumption being correspondingly increased.

The pressure readings and ratios of compression were as follows:—

		Abs. Press.		Compression ratio.
At beginning of 1st stage	=	12 lbs.	=	5·15
At end of 1st stage	=	61·8 ,,		
At beginning of 2nd stage	=	61·8 ,,	=	19·6
At end of 2nd stage	=	1,214·7 ,,		

In a well-designed compressor these two ratios should be equal, and in no case should the ratio of compression in any stage exceed nine.

For the above second stage ratio, the discharge temperature would be in the neighbourhood of 800° F., making the continued operation of valves and springs impossible and rendering the compressor unsafe to work. If anything, it is better to err by doing more work in the preliminary than in the final stages in order to avoid having the maximum temperature coinciding with the maximum pressure.

The following figures were taken by the author from the compressor of a M.A.N. submarine engine in 1912, when the engine was running at full speed.

			Abs. Press.		Compression. ratio.
1st stage	suction	=	10·3 lbs.	=	5
	delivery	=	51 ,,		
2nd stage	suction	=	51 ,,	=	17
	delivery	=	900 ,,		

Here again the staging is obviously wrong for this speed of the engine, and on examination the 2nd stage discharge valves showed signs of having been subjected to great heat, whilst all other valves were in perfect condition.

It should be observed that the compressors attached to marine Diesel engines are as a rule 20/30% larger than is necessary for the operation of the engine at full speed, the compressor being throttled to that extent on the first stage inlet. When the engine speed is reduced, less fuel is passed through the valve and consequently more injection air is passed into the cylinder. Also, the valve is open a longer time and this further increases the consumption of injection air, time being the more important factor in time area calculations.

This quantity is somewhat reduced by a natural fall in the injection pressure, which is desirable at low speeds for the smooth running of the engine. The compressor throttle is opened out full at these low speeds, and the ratios of compression then become equal as intended in the design. The engine, however, only runs at these low speeds at very infrequent intervals, and the writer pointed out at the time that the ratios of compression should be equal when running at full speed under the throttled condition and wrong when running at low speeds and full throttle, and this practice has now become general, but unfortunately, not before the two stage compressor had given way to the three.

To put the idea into practice in the design of a compressor let us assume that it is required to calculate the diameter of the K^{th} stage of a compressor having x number of stages.

Let D_1 = Diameter of 1st stage in inches.
C_1 = Clearance of 1st stage expressed as a fraction of the stroke.
C_k = Ditto for the K^{th} stage.
n = Exponent to the compression curve.
r = Compression ratio.

$$= \frac{\text{Blast Pressure lbs. per square inch abs.}}{\text{Pressure at beginning of 1st stage compression in lbs. per square inch abs.}}$$

Then diameter of K^{th} stage

$$= D_1 \left\{\frac{1}{r}\right\}^{\frac{k-1}{2x}} \left\{\frac{1 + C_1 - C_1 (r)^{\frac{1}{xn}}}{1 + C_k - C_k (r)^{\frac{1}{xn}}}\right\}$$

In calculating r, the value of the denominator was generally taken at 14·7. In the corrected form this should be reduced in proportion to the amount the compressor will be throttled when running at full speed. For example, if the compressor is required to give 1,000 lbs. blast pressure and to run with 25% throttling, 14·7 becomes about 11, and the correct value or r will be 92 instead of the old value of 69. This error of 50%

explains why the two stage compressor became discredited for marine Diesel engines.

The best solution, however, to the problem is to reduce the capacity of the compressor by the throttled amount and allow it to run at nearly full output for all speeds. For slow running, fuel valve lift reducing gear should be fitted in order to keep the ratio of amount of blast air to fuel oil as nearly constant as possible for all speeds. Such gear has the further advantage that much lower manœuvring speeds can be obtained, and no air injection marine Diesel engine should be considered complete without it. By this means continuous operation at piston speeds of 100/120 feet per minute are not uncommon, and this has not yet been bettered by any type of engine. This means that for oil engines of 4,000 H.P., 10 revolutions per minute or less can be anticipated, which is lower than there is any practical necessity for.

Neglecting the influence of valve seats and springs on the pressures has caused, and still causes, the under-estimation by as much as 40% of the running loads on all compressor parts. If allowance is made for this, and the same care is bestowed upon it as upon a working line of the engine, it can easily be made the most reliable part of any oil engine.

Increasing the number of stages is also advantageous, as apart from the greater efficiency of compression the piston clearances can be greatly increased, which is very desirable in the case of marine engines, without reducing the volumetric efficiency.

It may be broadly claimed that air injection gives a uniformly better fuel consumption than any other system and possesses a unique facility for efficiently consuming without special adjustment or preliminary heating almost every grade of fuel oil that can be picked up at any port. This elasticity is eminently desirable in the case of its marine applications, and is principally due to the fact that about eight times more energy is available for pulverising the fuel, than is possible with mechanical injection systems working at a pressure of 4,000 lbs. per sq. inch. For instance the *Camellaird-Fullagar* Opposed Piston Engine starts from all cold with oil as heavy as ·96 S.G., this being the heaviest so far tried under these conditions and quite a long run with oil containing as much as 4% ash has been made. This latter is not to be recommended for any engine, and is only quoted as showing the accommodating nature of air injection and what can be done in cases of necessity by some types of engines without detriment.

It is admitted that the engine would be simpler without the compressor, but when the many advantages it offers are taken into account, the added complication is fully compensated for. The increased mean pressure obtainable covers the cost of its addition to the engine with something to spare on the credit side.

We must, however, be sure that what is substituted is at least as good, if not better, than that which we wish to displace. It has yet to be proved that oil under pressure of 4,000/10,000 lbs. per square inch and red hot pistons is a step in the right direction and until this is done the wisest course appears to be to keep an open mind on the subject.

At present there is an endless variety of injection systems with oil pressures of varying amounts up to 10,000lbs. Some are cold starting, and most require preliminary heating with blow lamps, electricity or hot water in the jackets and pistons, whilst one starts with the ordinary electric spark until the bulb is hot.

For large marine engines, the favourite mechanical injection system consists in having a common pressure main in which the oil is kept at a steady pressure of 3,000/4,000 lbs. and to which all fuel valves are connected. In this application the fuel valves are under pressure all the time the engine is working, and should one hold up, as sometimes happens, a large quantity of fuel is at liberty to blow into the cylinder at any period of the cycle.

In a four-stroke engine the cycle requires 720° of crank angle to complete, whilst the fuel is admitted for less than 40°. This means that the pressure is only required for 1/18th of the time it is actually on the valve seat. Hence any leakage at the valve has ample time, especially in low speed engines, to assert itself, and if pre-ignitions are to be avoided, careful and frequent attention must be given to the valve seats. The effect on the fuel consumption is also bad, and the colour of the exhaust leaves much to be desired when one has got used to the absolute smokelessness of the air injection engine.

The writer devised a system to overcome this objection, and very promising results were obtained from the preliminary trials before the system of mechanical injection was abandoned for reasons not connected with this gear. Fig. 14 visualises the time the pressure is on the valve seat and the time the pressure is required to act, in a normal gear. The action of the new gear shown on Fig. 15 is such that the cylinder injection valve seat is relieved of all pressure from A to E and from F to D.

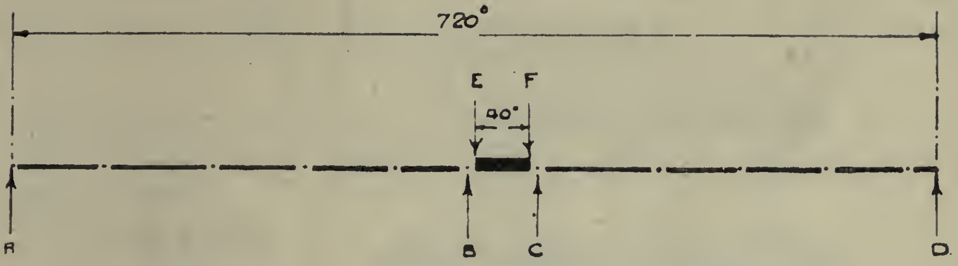


Fig. 14.

P is the pressure valve connected to the pressure main and E is an exhaust valve, both of these valves being opened and closed by the cam shown, and in such a manner that both valves are not open at the same time. Valve E is open from A to B, and from C to D, whilst valve P is only open from E to F, or during the time the oil is required in the cylinder. The short distances B E and F C, each representing about 3° of crank angle, are the necessary laps to ensure that valves P and E, Fig. 15, are not open at the same time. The fuel valve F in the cylinder is opened by the oil pressure acting on the underside of the piston formed on its spindle, and is spring closed. The oil displaced by valve F dropping on its seat is returned to the fuel oil drain tank which ensures rapid closure and stops dribbling at the jet and consequent after and inefficient burning of fuel in the cylinders.

In the ordinary system of mechanical injection, before the spring can close the fuel valve, it must push a volume of oil, equal to the area of the valve spindle multiplied by its lift, into the cylinder, or back down the pressure main. This is the reason why in some high speed engines the roller cannot follow the cam, the pressure holding the valve open after the cam has passed and inefficiently prolonging the injection period. The

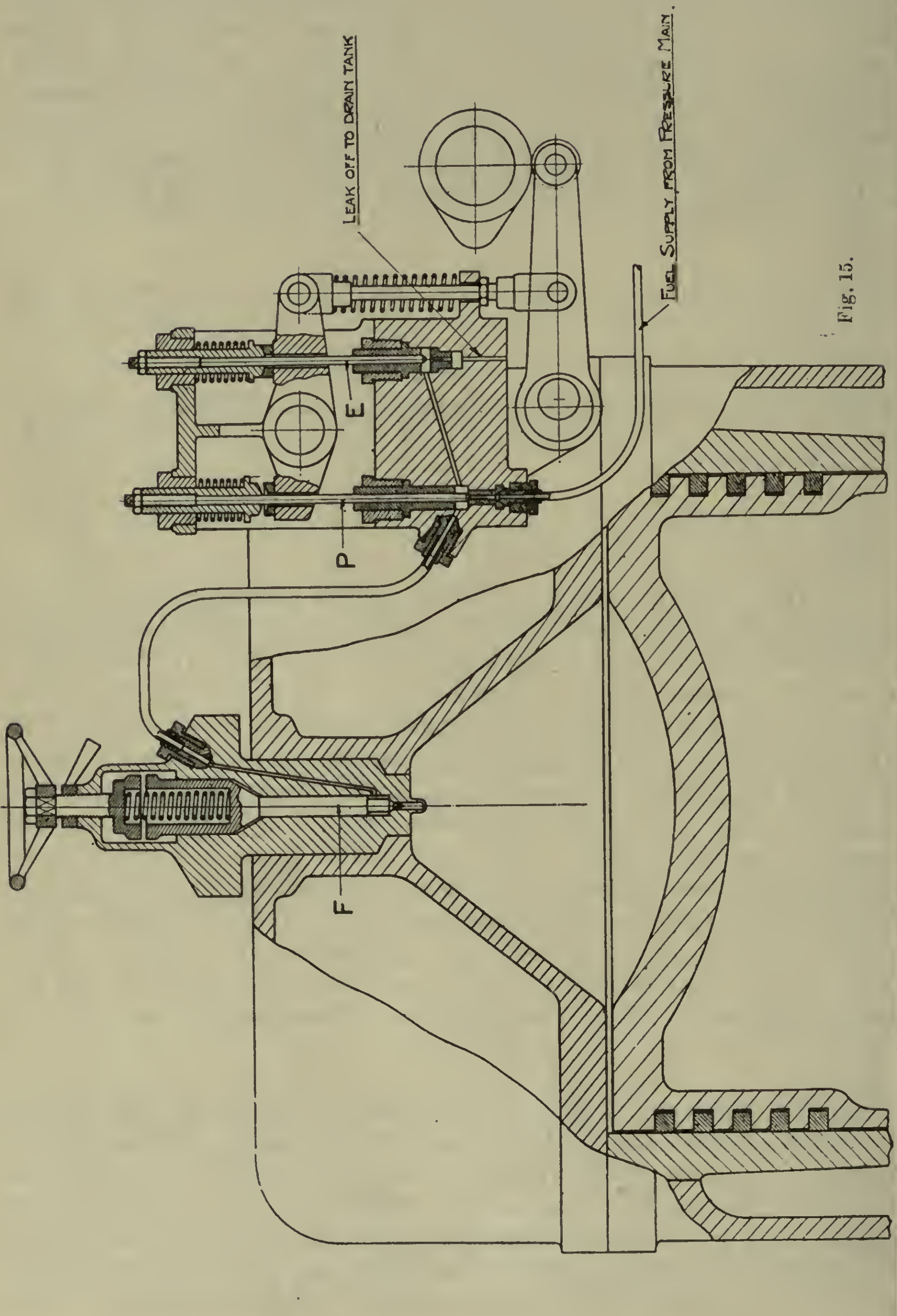


Fig. 15.

inertia of the valve parts is so small in proportion to the spring strength as to be almost inappreciable, and observations to the effect that this is the cause of the roller jumping are incorrect.

The high bearing pressures common to most marine Diesel engines are traceable to their land origin. The pressure of 2,000 lbs. per square inch on the gudgeon pin is a legacy from the trunk piston days when, owing to the confined space, a larger pin could not be put in. With the coming of the open crosshead type, however, there is no reason why this should have been overlooked and allowed to remain at such a high figure.

The following table of bearing pressures of various makes of engines has been compiled from published details and is instructive :—

	1	2	3	4	5
	Sulzer.	FIAT.	Fullagar.	Burmeister.	Werkspoor.
Gudgeon Pin	1850	1700	1350	2400	1900
Crankpin	1125	1290	720	1200	1060
Main Bearings ...	465	645	290	760	440

Engine No. 4 is a 1912 one.

The bearing pressures for marine steam engines is recommended not to exceed 250, 500 and 1,000 lbs. per square inch for main bearings, crankpins and gudgeon pins respectively. Such engines are, however, only drip lubricated and have reversible loads on pins, etc., which advantage is not possessed by engines marked 1 and 2 on the table. These two engine have bearing pressures much too high, considering the nature of the load, and considerable reductions will have to be made to fit them into correct marine practice.

Engine No. 3 is much better in this respect, the *Camellaird-Fullagar* cross rod principle making all loads reversible. In addition, the extremely low pressures throughout should be noted, this being more in accordance with sound marine practice.

Engines Nos. 4 and 5 are four-cycle, and although they have a much lower mean bearing pressure than 1 and 2, are too high for marine purposes. As a rule high bearing pressures mean severe stress conditions, and any increases in pins, etc., to bring these down would be a much needed improvement on an engine which has to stand the severe and hazardous duties of the sea.

It should be stated, however, that for equal maximum pressures, the mean bearing pressures for oil engines are about 20 per cent. less than for steam engines, and these are further relieved by the higher inertia pressures. Also, all oil engines as a rule are forced lubricated. These three factors will, therefore, ameliorate the conditions under which oil engines operate, but not sufficiently to make the bearing pressures at present in common use acceptable as sound engineering practice.

A Member: I quite agree with Mr. Shannon that the more failures brought to public notice the better it will be for all those connected with the manufacture of internal combustion engines, who ultimately, will arrive at something equally reliable to the steam engine. But why have the failures been hidden? The probability would seem to be that these failures have been so frequent or so bad that they were afraid to let them become known.

A Member: It would seem that the author has a prejudice against coal. Coal is the only thing we have in this country which has made the country what it is to-day. Both oil and coal are good fuels, and you must use either one or the other. Our friend does not take into consideration that in the use of this oil there is a dead loss in obtaining "Diesel" oil itself. A considerable amount of coal is necessary to distil that oil from the crude in order to make it into a condition of oil suitable for Diesel engines. Most of the Diesel engines do not use residual oil containing probably up to 20 per cent. of asphalt; but that is distilled, and in distilling it you take probably 10 per cent. of the fuel to do so. This oil with its asphalt content is the fuel used under steam boilers. In regard to Diesels, I think the price of this oil is £11, and heavy oil £8 per ton, and the author has quoted coal at £2 per ton. I think, with all due respect to him, that the shipowners will choose the best results as it appears from the point of view of touching their pockets. There is a paragraph in the paper about the utilisation of the distillates of coal. I should like to know what the author would do about that. Many of the coals of Great Britain would yield no by-products, and you might get a bucket of water if you distilled it. However, I do not wish to advocate a misuse of coal or even oil, but both must be closely classified for their particular purpose. Both are excellent fuels for their purpose of power generation, and finally I suggest that better results will be obtained if we tackle the subject more from the point of " The use of Oil and Coal as a fuel " rather than " Oil versus Coal as fuel."

A Member : Is it a fair and equitable distribution of pressure when we take these tabulated figures as given us, with the various pressures from gudgeon and crank pin to main bearing, or is that caused through what you might say a constant pressure engine, not as we have with the reciprocating engine?

The Chairman : The author has said a lot about compressors, and has told us that they never give much trouble. What oil engine experience I have had has proved that the compressors were the biggest trouble in the engine room. There was always one thing or another going wrong with them and certainly they gave more trouble than the main engine themselves. I am glad to hear that so many improvements have been made in them.

By correspondence :—

A Member : A considerable portion of Mr. Shannon's interesting paper has been devoted to an examination of the respective merits of mechanical v. air injector systems, and I am in complete agreement with the general conclusions he draws.

An air compressor which is well designed, gives no cause for anxiety, and is at least as reliable as, say the average air pump and condenser in a steamship. Good design, however, is not so dependent upon the compression ratio per stage, as he suggests, provided adequate and satisfactory cooling is provided for in the right place.

The M.A.N. compressor referred to by the author may have shown signs of heating at the discharge valves, but did these valves in fact cause any trouble? The exhaust valves in a motor-car engine or aero engine may work at a dull red heat with perfect satisfaction, and compressor valves are also entirely reliable working in temperatures far above that due to a 9/1 ratio, provided the means for cooling are propely designed.

This and a number of other problems connected with air compressor designs, have been thoroughly investigated by the leading Diesel engine manufacturers, and any firm of long experience will provide a compressor, the reliability of which has been proved beyond doubt.

The consumption of air at low speeds should, of course, be reduced and the only entirely satisfactory way of doing this is to reduce both the lift and period of opening of the fuel valve, as is done in all Sulzer engines. In this arrangement an extremely simple means is provided whereby both the lift and opening period of the fuel valve can be reduced from a maximum to no opening at all, while at the same time the point at which the fuel commences to be injected remains practically constant.

He makes some encouraging remarks about the two-cycle engine, until we reach the end of his paper, where he criticises the Sulzer and Fiat engines on the score of high bearing pressures. I cannot say whether the Fiat has had some unfortunate experience from the causes he suggests, but it is certainly not the case with Sulzer engines of the two-cycle type, experience of which extends to over 500,000 h.p.

The bearing pressures in Sulzer Marine Engines are much lower than those stated; the gudgeon pin for instance works at about 1,300 lbs. pressure, while the main bearings are capable of working quite satisfactorily even without forced lubrication, although naturally this is provided as it is far the best and easiest means of lubricating the various parts.

It surely is impossible to dogmatise on the question of what bearing pressures should be adopted. In a thrust block for instance, careful and scientific design has enabled much higher loads to be carried than the 50 or 60 lbs. per sq. in., which was common practice in the ordinary horseshoe block.

Certain general rules can be suggested, but they cannot cover all cases and take into account various refinements in design which materially affect the case. If a certain design gives complete satisfaction after many years experience, it must not be turned down because it does not comply with some simple general rules. In this and many other respects actual practical experience provides the best material for the basis of a design. With regard to the *Camellaird-Fullagar* cross-rod principle, is it not a fact that in this design the guides for the cross-rods are subjected to a continuous load always in the same direction, and consequently have a far more difficult duty to perform than the ordinary cross-head guides which have to deal with the intermittent loads due to the angle of the connecting rod? If it is possible to overcome by good design in a *Camellaird-Fullagar* engine troubles that might certainly be expected from an arrangement of this kind, it does not seem surprising that Messrs. Sulzers have also been able to solve the far simpler problem of a continuously loaded journal bearing.

The Author: There is no doubt that the steam engine has done a lot for us, and I should be the last one to say it has not. I entirely agree with the gentlemen in the remarks that have been made. The steam engine certainly won its place in history, and had it not been for the studies made of the steam engine, the oil engine would never have been where it is to-day. The oil engine is the steam engine with the oil cylinders put on top. I think I have a grumble when I say that the internal combustion engine might have taken up a little more attention in this country, when we consider that probably the greatest heat engine invention since the days of Watt, that is, the incorporation of compression before ignition was known in England for about twenty-five years before it was taken up in Germany, and immediately caused that bump in the curve.

A member stated that in those days little was known of the real nature of heat, which was very true, and it might also be very true to-day. We know a great deal more, but we have still a long way to go. The figure of £24,000 per ton which I gave as the value of a certain derivative that is contained in coal is correct. It certainly is a dazzling figure, but the derivatives of coal are extremely valuable, and some are worth more than the above figure. Probably most of the dyes and scents we buy in commerce are produced from coal tar; saccharine and aspirin are derivatives of coal. I do not say that when we buy a ton of coal we must expect a lot out of it. To get a ton of derivatives worth £24,000 you might have to distil a million tons. Take a country like Germany; there very little coal is destroyed in open fires. It is nearly all distilled in some form or other, and most of the heating of rooms is done by coke after all the gas has been taken out and the tar and all other things recovered; Germany depends on the distillation of coal to get the raw materials to keep many industries going.

A Member referred to the large amount of coal in this country that did not hold these valuable qualities. All coal contains gas; in fact it would not be coal if it didn't, and would therefore give derivatives, and the proportion of coal in this country that could not be efficiently distilled must be small. In regard to the reference to the failures of oil engines in the past, I know of many firms whose experiments in connection with oil engines would not be covered by £100,000. I believe Holzwarth has spent £50,000 on his gas turbine, and has not produced a workable machine; but for engineers who take a long view of the problem, the gas turbine does not appear quite hopeless. As long as there is a little hope in the problem I think a little encouragement from the State would do good, although I am of the opinion that individual effort is the best.

An idea originates in one head only, and if the community can help in developing that idea, I think good is done without destroying the individuality of the man.

The question of the absorption of heat does not depend upon the weight of metal. It is simply the question of the surface of the deck exposed to the sun and the surface of the keel exposed to the temperature, say twenty feet below the surface of the sea. I have not, however, made any calculations and perhaps I might as well ask you to invent a telescope that would enable you to see the Battle of Waterloo. Still, the idea is there, and something might be done when one considers that an average ship has about 16,000 sq. feet of deck surface.

A Member referred to the loss of heat in distilling oil. A great proportion of the oil used in Diesel engines is a by-product from the oil wells. That is, when the oil is taken from the ground it is distilled, the first thing to come off being ether. Following that we get petrol. Then they turn it over to another condenser, and the next thing is paraffin; and after that you might get gas oil and the next, I think, would be heavy Diesel oil, with a specific gravity of ·9 to ·93. After that the lubricating oils come. After the lubricating oils we get vaseline and fats.

The only use we have at present for the large fraction between gas and lubricating oils is to burn it in heavy oil engines or under boilers. The stated efficiency of the Diesel engine ignores the heat lost in distillation.

A Member: That is so.

The Author: In regard to Diesel engines, we have run the *Camellaird-Fullagar* on oil supplied to the Brocklebank Line for burning under the boilers. I do not say that it is a general practice, but the fact is there. A good deal of oil is burnt under boilers that could be three times better used in Diesel engines. A member raised the question of bearing pressure. I assumed a pressure of 500 lbs. in the cylinder in all cases so as to put all engines on the same basis. The areas of the bearing surfaces are therefore in inverse proportion to the pressures given in the table.

Engines with high pressures have small bearings and consequently high stresses. One speaker referred to the inertia of air. An air column has very serious inertia effects. In the exhaust pipe of a two-cycle engine, unless very carefully designed you can get a pressure of 7 lbs. at one end and a vacuum of 7 lbs. at the other. I think I have missed out many of the points that I should have replied to, but I cannot remember any more, and I have only noted down a few of the leading ones.

A Member: A Diesel ship could make a voyage of 24,000 miles without bunkering, whereas steamers would have to go out of their way frequently to get to coal ports. A steamer going East would probably coal five times between here and the Straits Settlements, and an oil ship could go there and come back without touching any port for oil supply, and, further, you have no ash to get rid of. You put the oil on board and it disappears. In regard to the *Aquitania*, I believe her engineers are highly delighted with the oil fuel she burns under her boilers. You can add their delights to the delights of the oil engineers, and you will have something to think about.

The Author: In answer to the correspondence, the M.A.N. compressor referred to has made a non-stop run of 48 hours. Undoubtedly the valves could run a long time but the high heat takes the temper out of the springs and trouble is almost certain.

As regards the high bearing pressure given in the table for the Sulzer engine, these were taken from the latest published drawing of this engine and the figures quoted show all the engines in their correct relative positions.

SOME OBSERVATIONS ON MARINE OIL ENGINES.

The pressures on the oblique rod guides of a *Camellaird-Fullagar* engine vary from a maximum of 50 lbs. per square inch down to 10 lbs. during the stroke, and although the direction does not alter, this problem cannot be considered in the same category as a gudgeon pin carrying 1,800 lbs. per square inch. I am glad to note that the correspondent is a two-cycle enthusiast, as time will prove that this will be the engine of the future.

THE CAMELLAIRD-FULLAGAR DIESEL ENGINE.

In this engine two cylinders placed together have to be adopted as a unit—as shown diagrammatically in Fig. 5a. Each cylinder has its own crank, a single one only, and the upper piston A of one cylinder is connected to the lower piston B of the second cylinder by diagonal rods, the upper piston C of the second cylinder being similarly connected to the lower piston D of the first cylinder. Thus, as the combustion takes place between the pistons A and D, D is pushed downwards and drives its crank round, the piston C, following through the action of the diagonal rods, while the piston A rises and draws the opposite piston B upwards. Thus, in place of a unit of one cylinder with two pistons, three connecting-rods and three cranks, we have a unit of two cylinders, with four pistons, but only two connecting-rods and two cranks, and with the addition of four diagonal rods, the unit giving double the power with the same size of cylinder. As far as working parts are concerned we have one rod and one crank per cylinder, as against three rods and three cranks, and so far the advantages are obviously in favour of the Fullagar engine, apart from the question of balance, which we shall deal with later. The argument cannot, however, be concluded quite so simply, for there is the effect of the diagonal rods to be considered, which at first sight looks much more serious than it actually is. The only possible objection which we can see to these diagonal rods is one which is common to all Diesel engine work, and that is the need for very accurate work in machining the face of the slippers on the end of the piston to take the lateral thrust, in boring the diagonal holes in the crosshead to take the rods, and in getting the exact length of the rods; while, from the point of view of design, it is essential that the guide should be stiff enough to resist the thrust. If these points are satisfactorily met—as, of course, they ought to be and can be—the

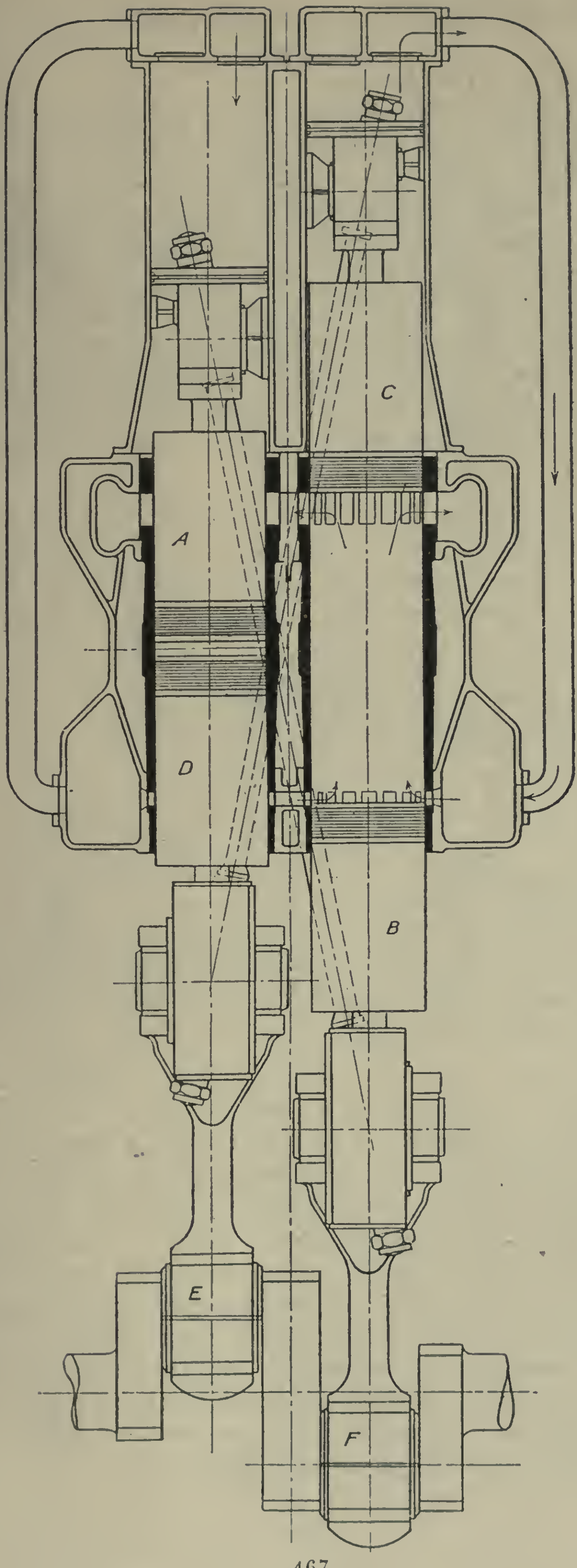


Diagram of the Cammellaird-Fullagar Engine,

Fig. 5A

drawbacks vanish, as the rods are always in tension, since when the piston A is rising, owing to the combustion pressure, the piston B is being pulled up against the compression pressure. It should be pointed out, too, that the thrust on the diagonal rod guides, unlike that of the ordinary connecting-rod guide, is at a maximum almost instantaneously just at the peak of the indicator card, and when the pistons are at their inmost position and the shoes thrust against the stiffest part of the guide. It should also be said that the obliquity of the rods is less than the maximum obliquity of the connecting-rod proper, so that the friction is less than of an ordinary connecting-rod in its guide.

One other point remains to be dealt with, that of balance. It will be seen that by means of the diagonal rods, not only are the total reciprocating weights exactly equal in amount for the two directions of reciprocation, and therefore balanced, but the centre of gravity of these parts has the same locus on a line between the centres of the two cylinders, so that the couple is also eliminated and only the angularity of the connecting-rod is left, which is, of course, dealt with in the engine of two units. The fact that there are only *two bearings for each pair of cranks makes for a further saving*, both in weight and manufacturing costs, though time will be required to show whether it will have any adverse effect on the life of the crank shaft, but as *the loads on the bearings are reversible, efficient lubrication can be ensured*. Fig. 9a shows the details of a two-unit engine. The cylinders are $18\frac{1}{2}$ins. diameter and the stroke of each piston 25ins. This engine gives 1,000 brake horse-power at 110 revolutions per minute.

As will be seen from the drawing, the two cylinders comprising a unit are bolted to a common cast iron plate which projects upwards above the ends of the cylinders to form a guide for the shoes to take the thrust of the upper ends of the diagonal rods, and the whole of this upper part is boxed in for a purpose which will be described later. The thrust of the lower ends of the rods is taken on a somewhat narrow plate projecting from the main connecting-rod guides through a white-metalled surface on the side of the main slipper, so that the guides are accessible for feeling on the outer sides. The lower part of the engine constitutes a very firm structure, the vertical columns being cast in pairs tied together, and there is not any sign of vibration on the starting platform.

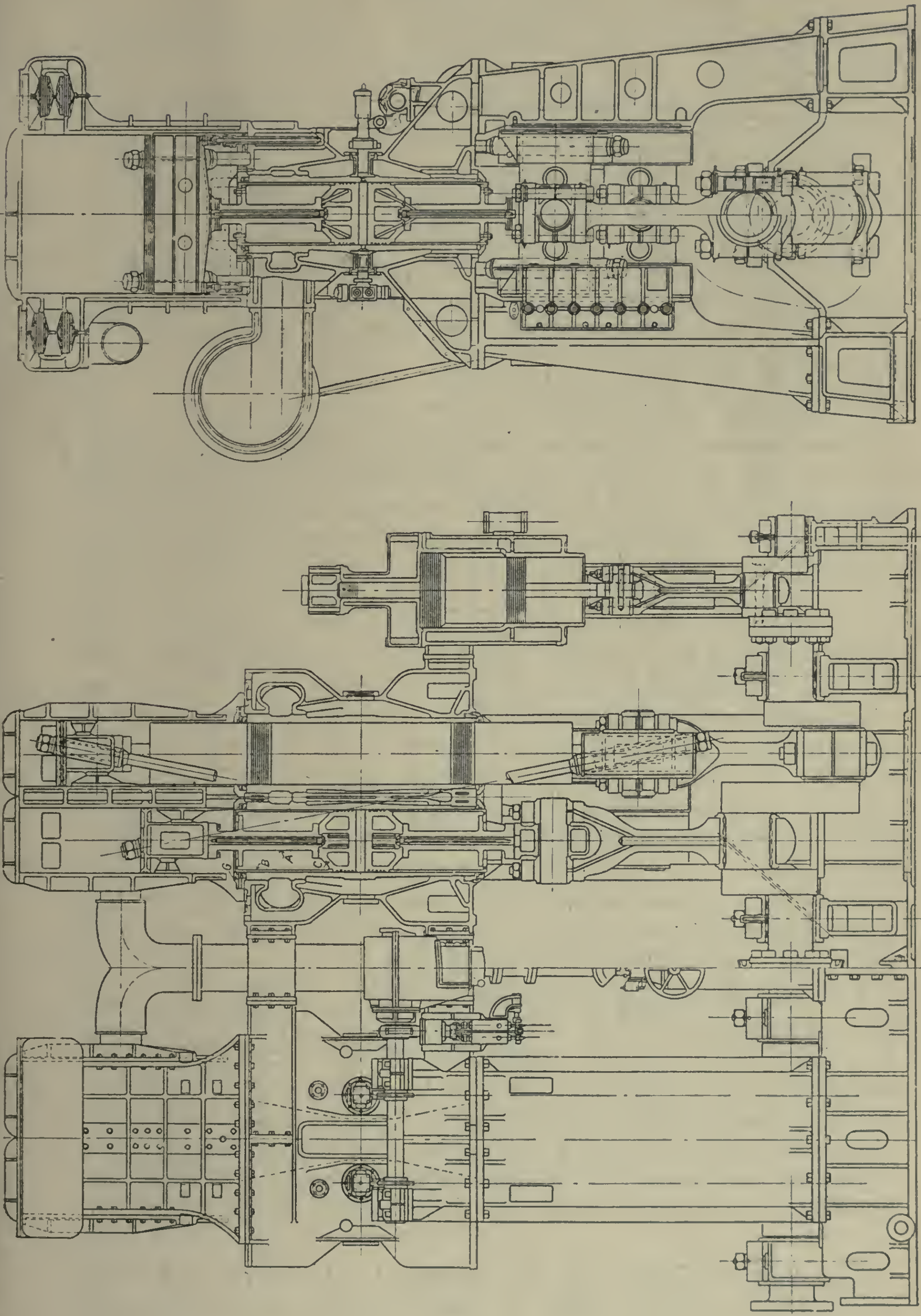


Fig. 9A, 1000 B.H.P. Two-unit Camellaird-Fullagar Diesel Marine Engine.

SOME OBSERVATIONS ON MARINE OIL ENGINES.

Twenty-four hour test of 1,000 b.h.p. *Camellaird-Fullagar* marine engine.

B.H.P., 1,000. I.H.P., 1,400.

Mechanical efficiency, 71·5.

Fuel consumption, ·391 lbs. per b.h.p. per hour.

Fuel consumption, ·28 lbs. per i.h.p. per hour.

Scavenging air pressure, 1¼ lbs. per square inch.

Blast air: first stage, 40 lbs. per square inch; second stage, 225 lbs. per square inch; third stage, 1,060 lbs. per square inch.

Lubricating oil pressure, 12½ lbs.

Circulating water pressure, 20 lbs.

Temperatures: circulating water inlet, 58° F.; circulating water outlet, 94° F.; lubricating oil drain, 94° F.; piston cooling water, upper, 102° F.; piston cooling water, lower, 66° F.

Fuel used, Anglo-American, sp. gr. 0·92; flash point, above 150° F.

The following is a copy of the report of tests made on the 1,000 b.h.p. engine in May, 1922:—

In accordance with your instructions I have carried out a series of tests of one of your *Camellaird-Fullagar* marine oil engines, and I have pleasure in sending you, along with this report a summary of all the principal data obtained during the tests, also a summary of the results of my calculations from these data.

The engine tested consists of two units combined on one crankshaft, and each unit of two power cylinders with four power pistons and two scavenge pumps.

The distinguishing feature of the engine is that the power pistons are connected to crossed rods which enable the number of cranks per unit to be reduced to two, thus shortening and simplifying the engine, as compared with other opposed piston engines and enabling it to run more steadily owing to the consequent great reduction of the unbalanced inertia forces. The cranks of each unit are inclined to each other at 180° and the cranks of one unit are inclined to the cranks of the other unit at 90°.

The power pistons are 18½ inches in diameter and their stroke is 25 inches. The scavenge pistons have an area of

921 square inches, and a stroke of 25 inches. All the auxiliaries, including the air compressor, the scavenge pumps and the circulating pumps, are driven by the engine.

The first series of tests were made during a continuous run of 26½ hours on the 20th and 21st of March, and the second tests were made during a run of eight hours on the 31st March. The arrangements made prior to the tests for obtaining all possible data regarding the operation of the engine were more complete than have ever been previously attempted in connection with an oil engine of so great a power as this, and, in spite of some difficulties, these arrangements proved satisfactory. In addition to the measurements usually made for testing an engine of this power, measurements were made of the rates of flow of the following:—

(1) The cooling water through the jackets of the power cylinders; (2) the cooling water through the jackets, intercoolers and after-cooler of the air compressor; (3) the water through the oil cooler; (4) the cooling oil for the power pistons; (5) the lubricating oil for the bearings; (6) the air to the scavenge pumps. In all these cases the circular orifice method, with tanks and stilling arrangements, was adopted. For the measurement of the pressure difference on the two sides of the air-measuring orifice a water column gauge was used and for the measurement of the pressure of the air in the scavenge belt a mercury column gauge was used.

Each set of indicator diagrams consists of 15 cards and all these were taken as nearly as possible at the same instant, but it was of course impossible to take the diagrams for the top and bottom pistons at the same time. All the indicators and thermometers used during the tests wers tested in my laboratory at the University, and the readings taken during the tests of the engine were corrected where necessary, in accordance with the calibrations of the instruments.

The fuel oil was weighed continuously on two weighing machines, and I tested these machines by means of standard weights and found that they were correct.

Samples of the fuel oil were taken during the tests and these were afterwards tested in a bomb calorimeter and in other ways, with the following results:—

	Oil Used in Tests.	
	Nos. 1 to 8.	Nos. 9 to 9a.
Calorific value (lower) B.T.U. ...	18,090	18,260
Specific gravity at 59° F.	0·878	0·880
Flash point	181° F.	185° F.
Viscosity at 62° F.	55 seconds	61 seconds

The calorific value of the sample taken for tests 9 and 9a was 0·9% greater than that of the samples collected for the other tests and it had a slightly higher density, flash point and viscosity.

I have calculated the value of n in the equation $PVn = C$ from cards taken from the power cylinders during test No. 9 with the following results, which are the means for four cards taken from each cylinder:—

No. of Cylinder.	Values of n. Compression.	Values of n. Expansion.
1	1·396	1·167
2	1·371	1·207
3	1·344	1·183
4	1·378	1·244

I have divided the test made on March 31st, into two, No. 9 and 9a, because the engine did not give as good results during the first four hours as it did during the last three hours.

I was informed at the time that this was probably due to boiler fuel oil having been used in the engine on the previous day.

The accompanying sheets contain the means of all the readings taken during the tests and the results derived from these data.

The results given on sheet 3 show that the power developed in cylinder No. 3 was consistently lower than that developed in the other cylinders.

We obtain for test 9a the following results, the percentages being based on the heat due to the fuel oil used per minute:—

Heat due to fuel oil used per minute 145,800 B.T.U.

	B.T.U.	%
Heat equivalent of indicated work per minute	64,730	44·4
Heat carried away by jacket water per minute	30,630	21·0
Heat carried away piston cooling oil per minute	5,230	3·6
Heat carried away to exhaust per minute ...	36,200	24·8
Heat lost by radiation, etc. (by difference) per minute	9,010	6·2
	145,800	100·0

From this we see that the brake thermal efficiency is 32·7%.

The distribution of the heat in the engine is shown graphically in two ways on sheet No. 7.

A set of indicator diagrams is given on sheet No. 5.

Various curves giving the results of the tests made in accordance with the propeller law are given on sheet No. 6 and on the same sheet the results are given on a brake horse-power base.

The weight of fuel oil used per b.h.p. hour in my tests (1) and (2) is approximately the same as that given in the paper read by Mr. James Richardson before the Institution of Naval Architects in April of this year for the Beardmore-Tosi engine of the motor ship *Pinzon.* This engine is of the four-stroke cycle type, with six cylinders, and its nominal power is 25% greater than the normal power of your engine.

During the whole of my tests the engine worked as smoothly and as noiselessly as any marine steam engine, the amount of attention required for its operation was almost negligibly small, the exhaust gases were absolutely invisible during most of the tests, and in no case were they more than barely visible.

The amount of lubricating oil used for the engine pistons was remarkably small, being only 0·164 gallon per hour when the engine was giving 1,115 b.h.p. This is equivalent to 0·147 gallon per hour per 1,000 b.h.p., or to approximately 7,000 b.h.p. hours per gallon.

At the conclusion of my tests a series of manœuvring trials were made, and the time for reversal from full speed ahead was 2½ seconds, firing in the cylinders began 2½ seconds later, and the engine was running full speed astern in eight seconds from the instant that I gave the signal to reverse.

The engine was run for some time at a speed of 16½ revolution per minute, firing taking place regularly in all cylinders during this time. This speed corresponds to a piston speed of 68·7ft. per minute, which is, I believe, lower than has ever been previously achieved in any internal combustion engine.

The design of the engine is simple, direct and free from unreliable mechanisms, and both its design and construction are in accordance with the great reputation established by your firm, and I believe that there is a great future for this engine.

(Signed) W. H. WATKINSON.

Tests of Cammell-Laird Fullagar Marine Oil Engine.

SUMMARY OF DATA RELATING TO TEMPERATURES.

	Date:—	March 20th	March 21st				March 31st
1.	Barometer Hg.	29.95	29.97	29.99	30.00	30.02	29.44
2.	Hygro-meter { Dry bulb	46.6	45.5	—	45.5	45.6	44.7
3.	Wet bulb	42.5	40.5	—	41.0	41.5	—
4.	Humidity of the atmosphere	.72	.690	—	.699	.720	—
	AIR COMPRESSOR AIR TEMPERATURES. FAH°						
5.	1st stage suction	100.0	101.3	81.8	80.1	87.8	103.4
6.	1st cooler, before	186.5	182.5	161.5	160.5	164.0	187.7
7.	1st stage cooler, after	137.0	132.5	122.0	114.0	119.0	134.8
8.	2nd stage cooler, before (1)	273.9	297.5	248.4	252.9	273.2	306.1
9.	2nd stage cooler, before (2)	168.1	164.8	132.0	135.0	146.0	174.4
10.	2nd stage cooler, after	91.0	88.0	64.5	77.0	81.0	93.6
11.	3rd stage cooler, before	273.2	282.2	191.3	217.4	244.4	282.7
12.	3rd stage cooler, after	136.2	135.5	96.5	104.0	118.0	135.7
	SCAVENGE AIR TEMPERATURES. FAH°						
13.	Inlet	41.0	41.1	41.7	41.9	41.0	47.6
14.	Outlet	81.7	78.8	60.4	60.8	65.3	86.0
	EXHAUST PRODUCTS TEMPERATURES. FAH°						
15.	From Cylinder No. 1	417	554	244.9	306	356	—
16.	From Cylinder No. 2	481	581	251	311	346	—
17.	From Cylinder No. 3	449	473	221	293	378	—
18.	From Cylinder No. 4	485	—	244.9	296	419	—
19.	Common Exhaust	525	590	263	338	419	462.5
	CIRCULATING WATER TEMPERATURES. FAH°						
20.	Common inlet	60.1	57.8	55.0	55.4	59.0	55.4
21.	Compressor water outlet	74.7	72.0	64.0	65.0	69.0	71.1

OUTLET TEMPERATURES. FAH^c OF COOLING WATER FOR JACKETS OF POWER CYLINDERS.

	Date :--	March 20th	March 21st				March 31st
1.	No. 1 Cylinder	104.5	109.4	89.6	97.7	103.1	105.0
2.	No. 2 Cylinder	109.2	113.4	95.0	98.6	107.6	106.5
3.	No. 3 Cylinder	109.6	108.5	96.8	102.2	110.3	100.7
4.	No. 4 Cylinder	110.7	125.6	92.7	103.1	118.4	108.7
5.	Mean temperatures, outlets	108.5	114.2	93.5	100.4	109.8	105.2
6.	Mean temperatures, inlets	60.1	57.8	55.0	55.4	59.0	55.4
	OIL COOLER TEMPERATURES.						
7.	Water inlet	61.7	58.6	55.4	55.4	58.9	57.6
8.	Water outlet	73.4	72.5	69.2	67.0	68.5	80.5
9.	Oil inlet	120.3	122.8	110.0	103.5	106.5	101.1
10.	Oil outlet	109.2	110.3	101.5	96.2	98.2	93.4
	PISTON COOLING OIL TEMPERATURES (OUTLET).						
11.	No. 1 Top piston, outlet F°	144.7	147.0	—	114.0	124.0	135.0
12.	No. 1 Bottom piston ,, ...	138.7	143.0	115.0	—	123.0	113.9
13.	No. 2 Top piston ,, ...	138.7	142.7	—	114.8	120.2	122.1
14.	No. 2 Bottom piston ,, ...	139.6	140.9	117.8	—	140.9	115.6
15.	No. 3 Top piston ,, ...	146.1	153.5	—	116.2	128.3	129.2
16.	No. 3 Bottom piston ,, ...	138.1	136.5	115.7	—	119.5	115.3
17.	No. 4 Top piston ,, ...	139.1	144.2	—	114.0	122.0	129.3
18.	No. 4 Bottom piston ,, ...	139.6	146.5	118.0	—	123.0	117.5
19.	Mean temperatures	140.4	144.6	116.6	114.7	125.1	122.2
	QUANTITIES OF COOLING WATER AND COOLING OIL.						
20.	Power cylinders jackets, pounds per minute ...	618	675	364	338	488	615
21.	Piston cooling oil per minute, pounds	—	328	248	316	389	*378
22.	Air compressor jacket and coolers, lbs. per min.	282	284	280	256	300	323
23.	Oil cooler water, pounds per minute	362	366	355	320	357	345
24.	Bearings Lubricating oil per minute, pounds ...	114	130	102	113	115	—

*Estimated from previous test results.

AIR COMPRESSOR—PRESSURE IN INTERCOOLERS, POUNDS PER SQUARE INCH.

25.	Stage 1	36	36	37	37	37	32
26.	Stage 2	167	162	187	160	152	177
27.	Blast	907	955	700	760	895	996

SUMMARY OF DATA OBTAINED FROM THE INDICATOR DIAGRAMS.

Power Cylinders—Mean effective pressures in pounds per square inch.

		No. of Cyl.														
1.	... ...	1	T.	86.1	92.0	96.3	101.2	52.2	57.9	56.8	60.0	77.8	80.4	85.8	92.3	For Power
2.	... ...		B.	97.8		106.2		63.6		63.2		83.0		98.8		Cylinders
3.	... ...	2	T.	86.5	95.2	100.0	109.0	50.7	61.6	65.5	72.7	74.8	83.9	84.5	95.7	I.H.P.=P_mR C.
4.	... ...		B.	103.9		118.0		72.5		80.0		93.0		107.0		where R is the
5.	... ...	3	T.	68.9	78.1	79.5	92.1	50.2	55.4	54.0	60.7	66.8	76.1	80.3	87.5	revolutions per
6.	... ...		B.	87.4		104.8		60.7		67.5		85.5		94.7		minute and
7.	... ...	4	T.	83.5	90.7	98.8	107.4	57.7	63.1	68.0	73.2	75.3	85.6	86.3	95.9	C=0.1358.
8.	... ...		B.	97.9		116.0		68.5		78.5		96.0		105.6		
9.	M.E.P. for all pistons			89.0		102.4		59.5		66.6		81.5		92.8		
10.	Revolutions per minute			117.3		118.3		58.5		90.5		110.6		121.1		
11.	I.H.P. Total			1420		1648		473		818		1220		1526		

Scavenge Cylinders—Mean effective pressures in pounds per square inch.

12.	... ...	1	3.14	3.26	0.78	—	2.56	2.56	For Scavenge
13.	... ...	2	2.34	2.44	0.41	—	1.90	3.47	Cylinders
14.	... .	3	3.00	3.14	0.83	—	2.46	3.24	I.H.P.=P_mR C.
15.	... ...	4	2.95	3.15	0.86	—	2.32	3.18	where C=0.2326
16.	M.E.P. for all pistons		2.86	3.00	0.72	—	2.31	3.11	
17.	I.H.P. Total		78.2	82.5	9.8	—	59.5	87.6	

Pressure of air in air belts of power cylinders in pounds per square inch.

18.	... ...	1.65	1.86	0.5	0.99	1.47	1.91

Air Compressor—Mean effective pressures in pounds per square inch.

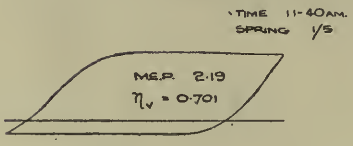
19.	... ... 1st stage	14.2	12.8	19.9	12.8	11.3	12.8	I.H.P. = 0.0169 P_mR.
20.	... ... 2nd stage	71.1	70.8	87.2	65.5	59.8	83.3	I.H.P. = 0.00875 P_mR.
21.	... ... 3rd stage	350.4	344.0	279.0	273.8	295.0	358.5	I.H.P. = 0.000764 P_mR.
22.	... ... I.H.P. 1st stage	28.2	25.6	19.7	19.6	21.1	26.2	
23.	... ... I.H.P. 2nd stage	31.3	31.4	19.1	22.2	24.8	37.8	
24.	... ... I.H.P. 3rd stage	31.4	31.1	12.5	18.9	24.9	33.2	
25.	... ... I.H.P. TOTAL	90.9	88.1	51.3	60.7	70.8	97.2	
26.	Power cylinder I.H.P. less compressor I.H.P.	1329	1560	422	757	1149	1429	
27.	Net I.H.P. of engine, i.e., I.H.P. of power cylinders less the I.H.P. expended in scavenge cylinders and compressor ...	1251	1477	412	—	1090	1341	

SUMMARY OF POWERS, CONSUMPTIONS AND EFFICIENCIES.

1.	Date of test	20/3/22 to 21/3/22.				March 31st, 1922.	
2.	Duration of test, in minutes	375	36	49	28	19	180
3.	Revolutions per minute (R)	117.3	118.3	58.5	90.5	110.6	121.1
4.	Mean effective pressure, power cylinders, pounds/sq. in.	89.0	102.4	59.5	66.6	81.5	92.8
5.	Brake mean effective pressure, power cylinders, pounds/sq. in.	64.97	76.8	36.29	46.62	57.86	68.48
6.	Power cylinders $(I.H.P.)_p$ (Sheet 3 line 11) ...	1420	1648	473	818	1220	1526
7.	Scavenge pumps $(I.H.P.)_s$ (Sheet 3 line 17) ...	78.2	82.5	9.8	—	59.5	87.6
8.	Air Compressor $(I.H.P.)_c$ (Sheet 3 line 25) ...	90.9	88.1	51.3	60.7	70.8	97.2
9.	Brake Load in pounds (L)	3093	3653	1722	2210	2753	3254
10.	Brake horse power	1037	1235	288	571.4	870	1126 B.H.P $= \frac{L R}{350}$
11.	Mechanical efficiency = B.H.P./$(I.H.P.)_p$ ...	0.73	0.75	0.61	0.70	0.71	0.738
12.	,, ,, = B.P.H./$(I H.P)_p$ — $(I.H.P.)_c$	0.78	0.79	0.68	0.75	0.76	0.79
13.	,, ,, = B.H.P / $(I H.P.)_p$ — $(I.H.P.)_c$ — $(I.H.P.)_s$	0.83	0.83	0.70	—	0.80	0.84
14.	Weight of fuel oil used per hour, lbs.	434	540	137.8	244.3	354	479.2
15.	,, ,, ,, B.H.P. hour, lbs....	0.418	0.436	0.478	0.428	0.407	0.425
16.	,, ,, ,, $(I.H.P.)_p$ hour, lbs...	0.305	0.327	0.290	0.298	0.290	0.314
17.	Thermal efficiency	0.462	0.431	0.486	0.473	0.486	0.444

SHEET 5.

CARDS FOR
SCAVENGE CYLINDERS.



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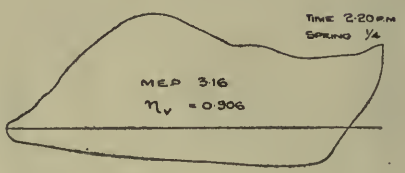

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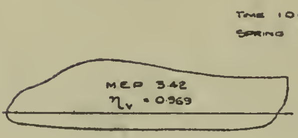
NOTE.- η_v DENOTES VOLUMETRIC EFFICIENCY



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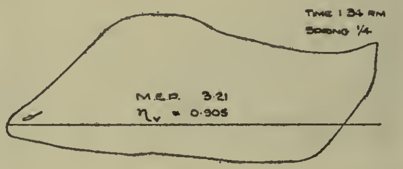

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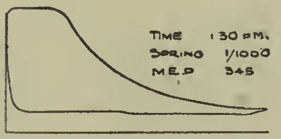
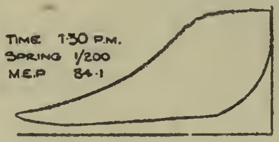
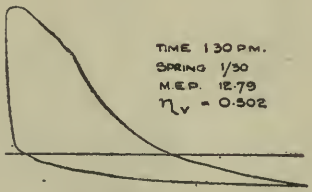


H.P.



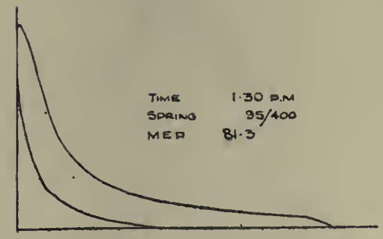
I.P.



L.P.

AIR COMPRESSOR CARDS.

TYPICAL INDICATOR DIAGRAMS FOR POWER CYLINDERS.



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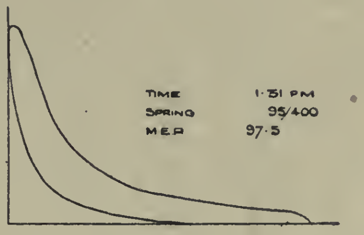
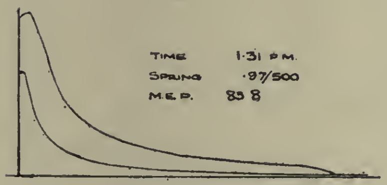


BOTTOM.

Nº 1 CYLINDER.



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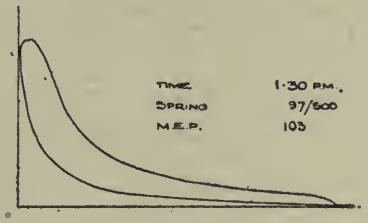

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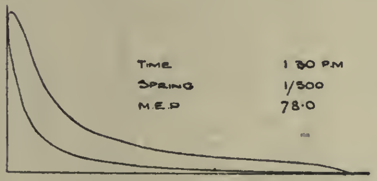
Nº 2 CYLINDER.



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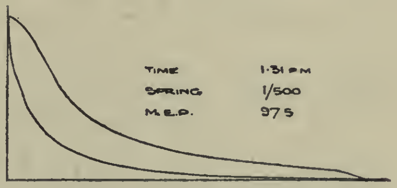

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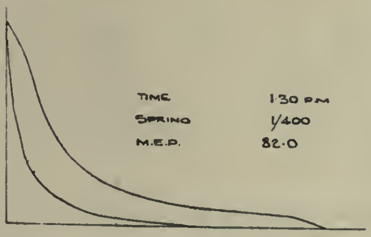
Nº 3 CYLINDER



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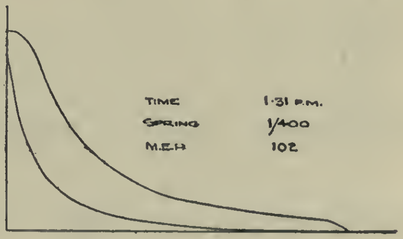

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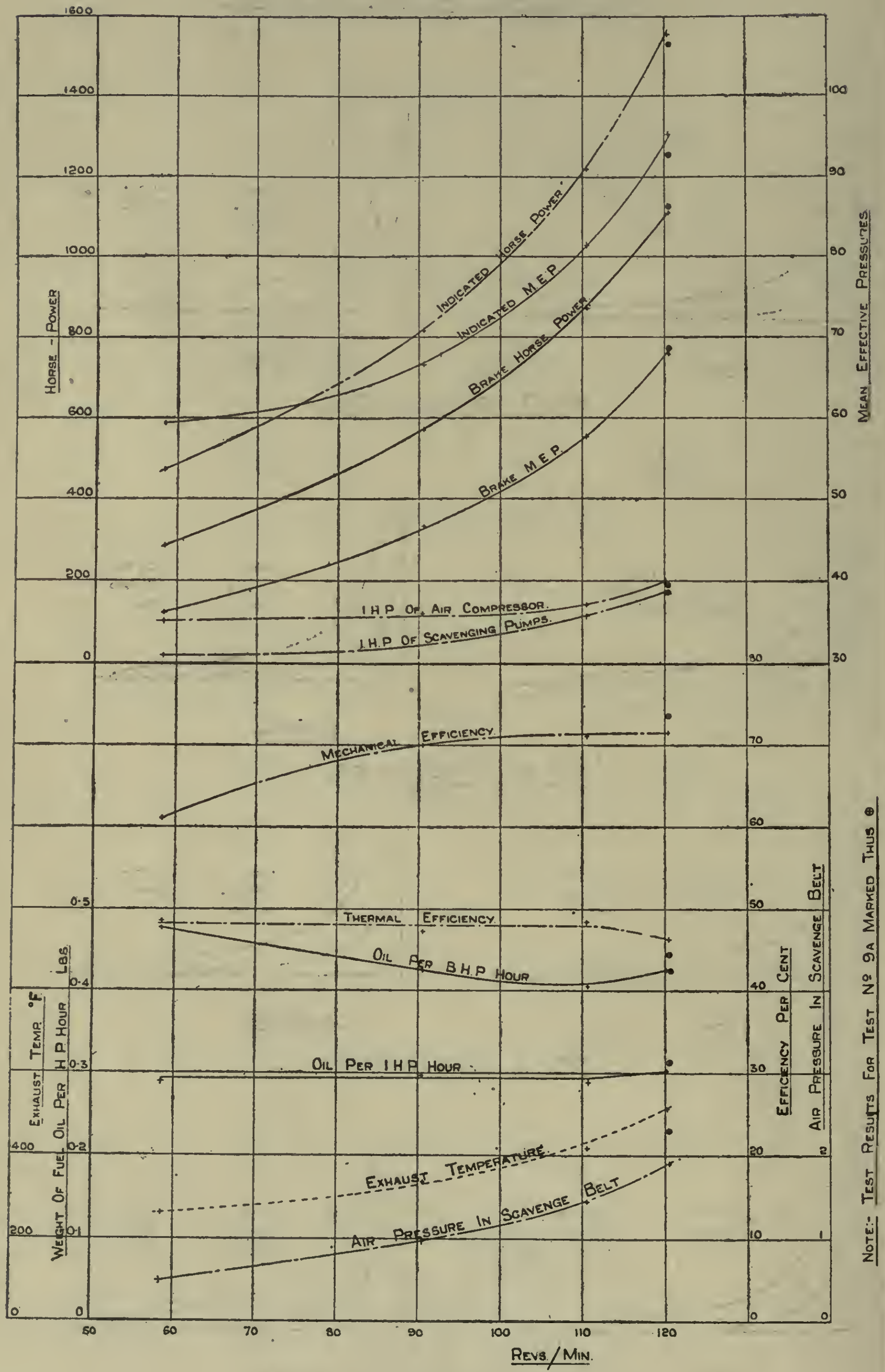
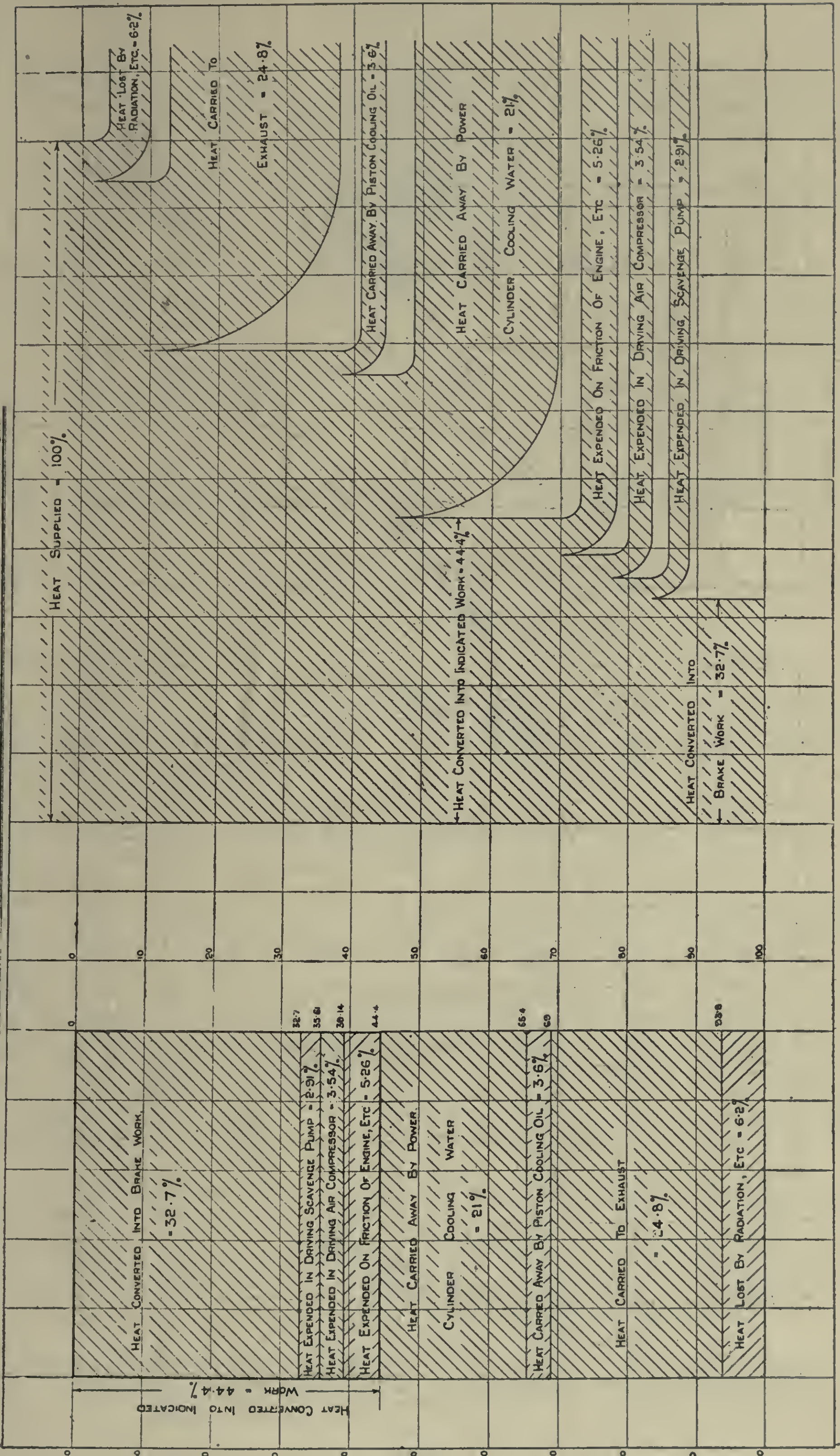
Nº 4 CYLINDER

SOME OBSERVATIONS ON MARINE OIL ENGINES.



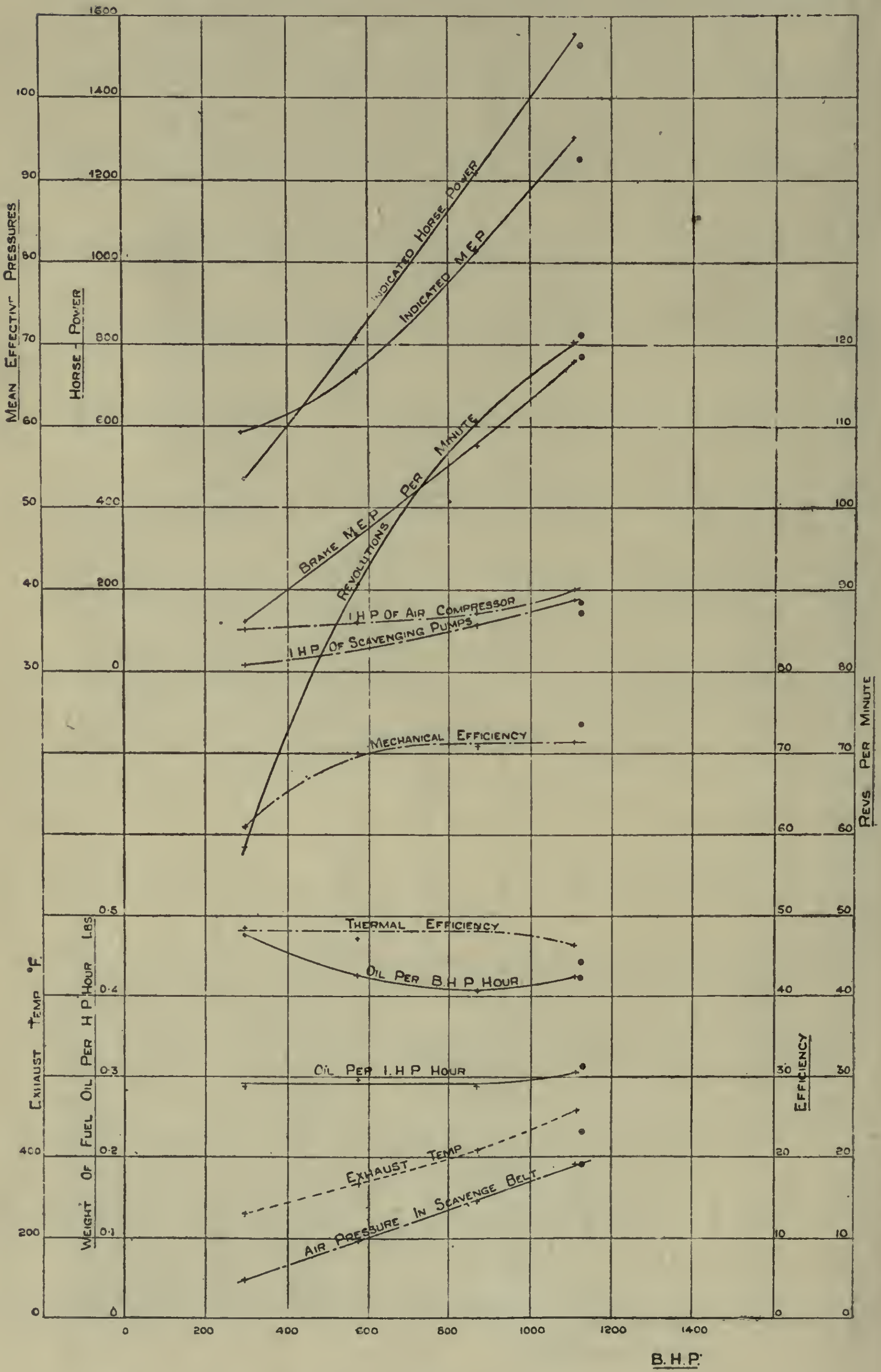


Sheet 7.

Diagrams Showing The Distribution Of Heat - Test 9A

SOME OBSERVATIONS ON MARINE OIL ENGINES.

SHEET 6.



SESSION 1921-22.

President: SIR JOSEPH P. MACLAY, BART., LL.D.

Auxiliary Machinery of Oil-Engined Vessels.

BY MR. WALTER POLLOCK (Member).

Read Tuesday, March **22, 1921.**

THE problem of auxiliary machinery presents greater difficulties with oil-engined vessels than with steamers. With the latter, either the donkey boiler or the main boiler supplies all the steam necessary for the capstan right aft, the windlass right forward and all the other auxiliary machinery in between. When one thinks of the price of coal in these days, and the appalling inefficiency of steam winches, the steam windlass, steam steering gear, and the steam capstan, etc., one wonders why such extravagance is still permitted. The simplicity of the steam auxiliaries and their reliability is probably the reason, but when the waste due to condensation in the long lengths of steam pipe and the general wastefulness of the average steam winch are taken into consideration, it surely only requires a well tested oil engine that can work in the open and prove itself reliable to make steam auxiliaries obsolete.

A great deal has been written of the advantage of oil-engined vessels over those fitted with steam engines, but one does not often read of the large amount of auxiliary machinery to be found in a large oil-engined ship, particularly air compressors; however, this point will be discussed more fully later.

We should first consider the alternative methods of powering the auxiliaries, then the methods available, together with the

AUXILIARY MACHINERY OF OIL-ENGINED VESSELS.

advantages and disadvantages and notes concerning their installation and working. The powering methods on all vessels must depend upon coal or oil, and the author will endeavour to show that the latter is much more economical and with proper engineers, the best makes of oil engines are equally reliable.

Much more depends on the training and instruction of the engineers and men in charge of the engine-room and the deck auxiliaries, than the majority of owners and superintendents are aware of. The explanation is, that too much is said of the simplicity of the oil engine, especially of the surface ignition or hot bulb type, which is usually adopted for oil engine driven winches, etc. Too little is said of the difficulty of getting trained men who really know their job, and of the almost impossible task of finding the stevedores' men able to properly manage the winches. There is no difficulty in finding dozens of men who can work steam winches.

Powering by steam means of course a donkey boiler and should this method be adopted, even with an oil-fired boiler, it should be fitted in a separate engine-room for reasons of cleanliness. The only advantage of the donkey boiler is that anybody can work it, and there is considerable flexibility in the steam winches and auxiliaries. The advantages of the surface ignition or hot bulb oil engine are that it can use cheap fuel oil, is very economical to run, has no loss by condensation in steam pipes, has no stand-by losses, the fuel is handled more easily than coal, and the whole installation with oil engines is about one-third the weight of the steam plant, this latter point increasing the amount of cargo carried and freight earned.

Oil engines using paraffin and/or petrol for fuel, are only used for small auxiliaries. Petrol is usually dangerous if used on board a ship and paraffin is more expensive than the fuel oils used by surface ignition or hot bulb oil engines.

If powering by oil engine is adopted it means:—

(*a*) A separate engine for each machine, or

(*b*) A generating station in the main or auxiliary engine-room for the whole ship, or

(*c*) A combination of (*a*) and (*b*).

AUXILIARY MACHINERY OF OIL-ENGINED VESSELS.

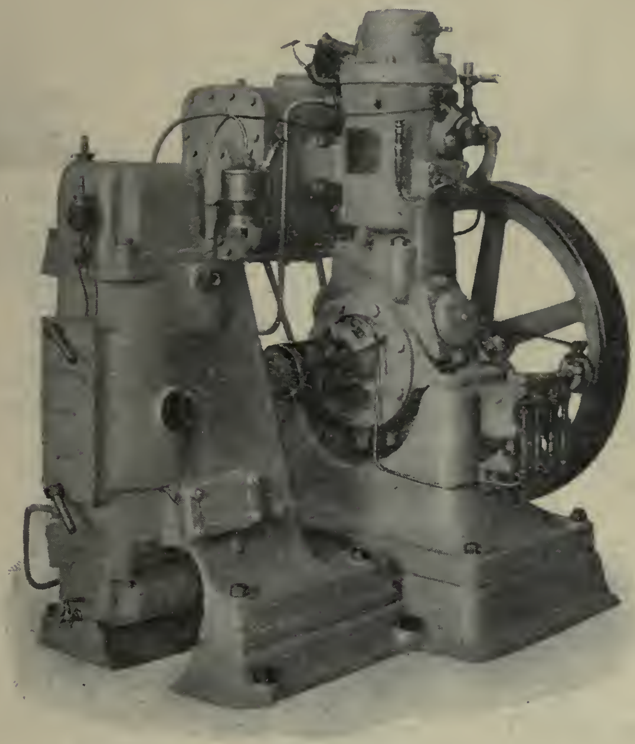


Fig. 1. Oil Engine and Air Compressor Set.

The separate engine is practicable for each air compressor, winch, capstan, windlass, pump and the electric light, but not so convenient for the steering gear. A verv efficient and economical method of driving the winches is to have an oil engine in a deck house between the hatchways, suitable for driving say four or six winches at a time, as shown in Fig. 2.

FUEL COSTS OF OIL AND STEAM WINCHES.

One Ton Steam Winch.	*One Ton Bolinder-Reid Winch.*
10 h.p. Engine.	8 b.h.p. Bolinder Engine.
Coal per week of 50 hours at say $4\frac{1}{2}$ lb. per h.p. per hour, plus $1\frac{1}{2}$ lb. for condensation = 3,000 lb.	Fuel per week of 50 hours at 6 lb. per hour = 300 lb. per week.
Say one ton, 7 cwts. of coal per week at 77s. 6d. per ton = £5 3s. 6d.	Say 34 gallons costing 1s. 4d. per gallon = say £2 5s. 0d. per week.

showing a saving of £2 18s. 6d. per 50 hour week in favour of the oil engine driven winch. As previously mentioned, the first cost of the steam winch is less than that of an oil engine

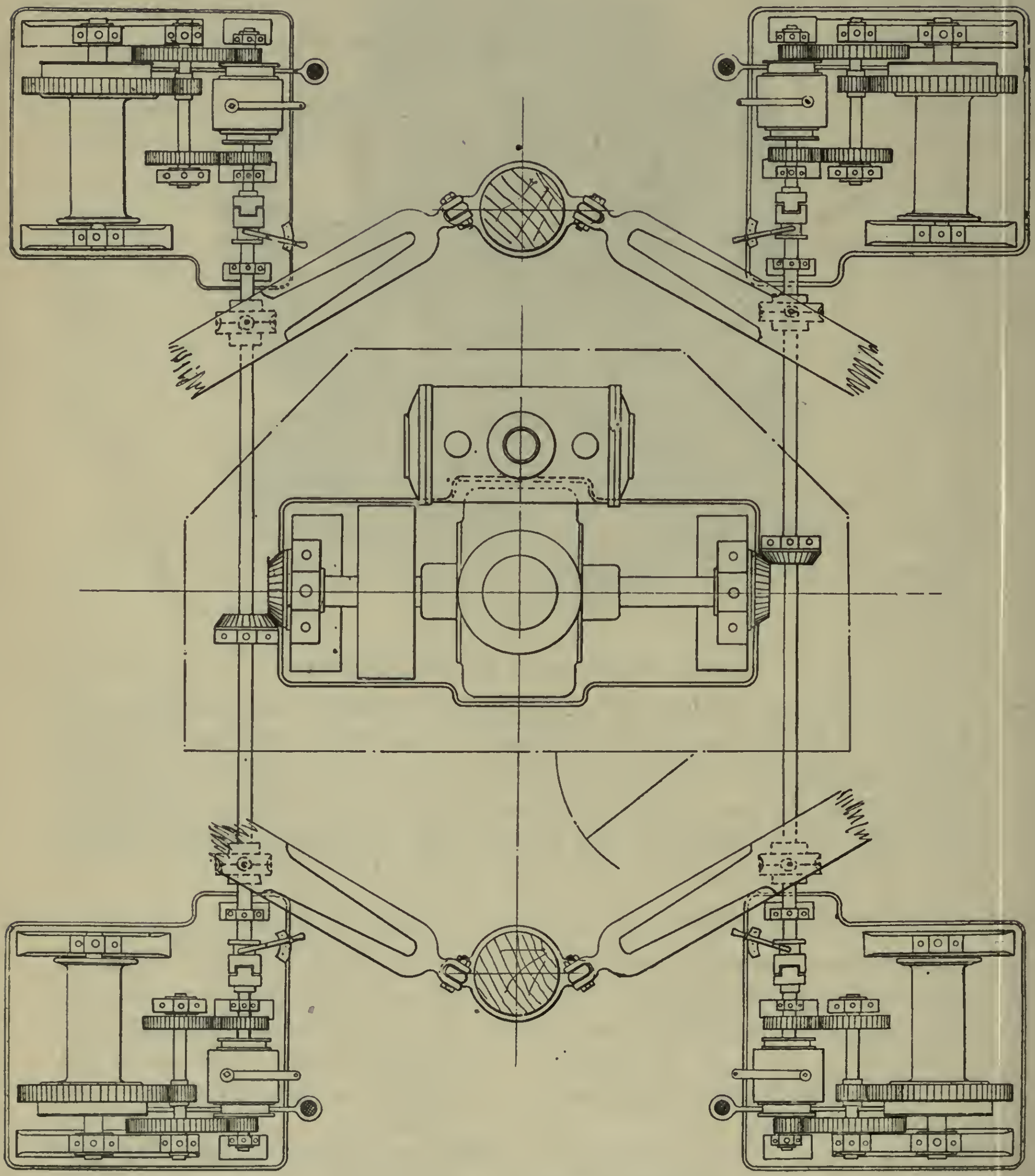


Fig. 2. Winches and Oil Engine.

AUXILIARY MACHINERY OF OIL-ENGINED VESSELS.

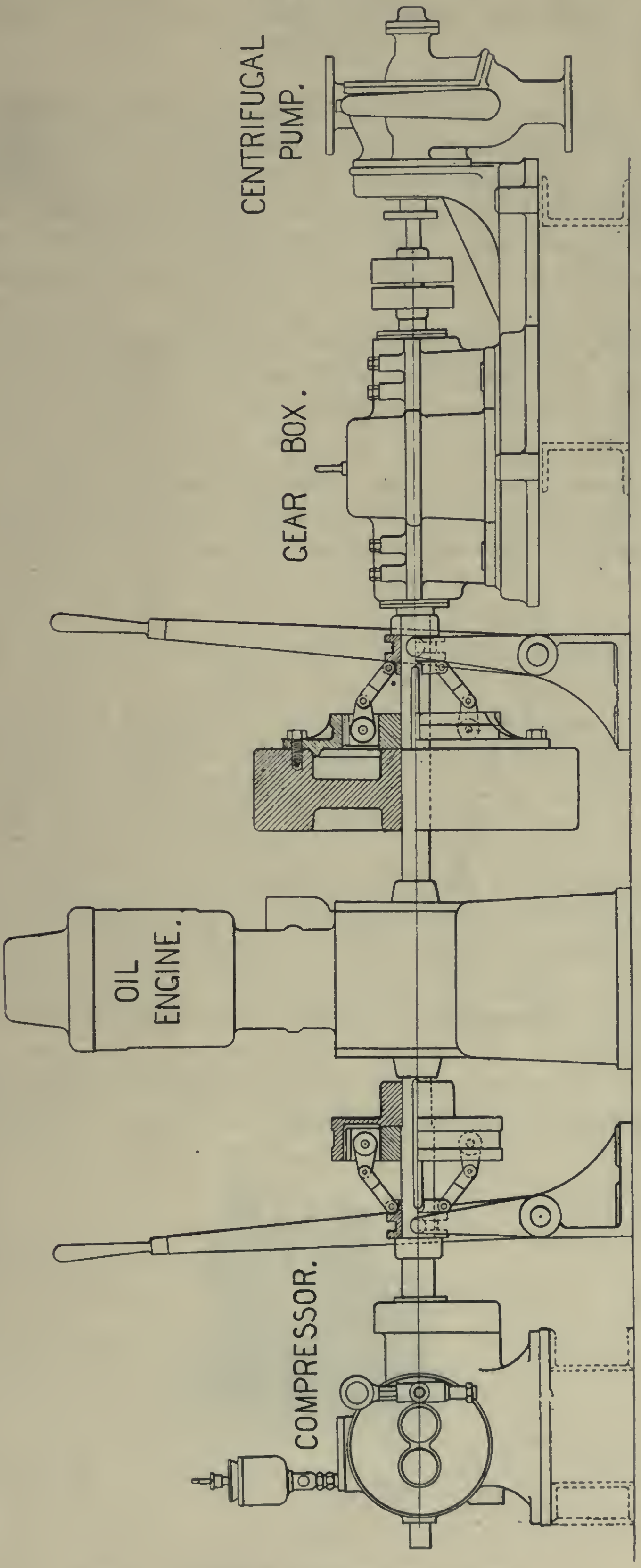


Fig. 3. Oil Engine, Compressor and Pump.

driven winch, but in addition to the saving by fitting the latter, a considerable saving is also effected by the abolition of the boiler with its consequent stand-by losses. It should be borne in mind that no packing of glands is required on the oil engines, whereas steam winches require a fair amount of attention in this respect if they are to be maintained in an efficient condition.

A central generating station can power the whole of the auxiliaries by supplying electricity or compressed air; the former is a strong favourite, although the latter could in some cases be used with advantage, especially for the steering gear.

The advantages of the "separate engine" are, the well known individual unit, the absence of stand-by losses and the highest possible economy of fuel, as only the power actually required is used and on the spot where wanted. The disadvantages are the greater initial cost of the large number of oil engines required and the increase in cost of installation due to the necessary water cooling connections.

At the present time a two ton oil-engined winch costs twice as much as a two ton steam winch. As more experience is obtained with the former, and as the demand and production increase, the cost will be reduced, although an oil engine will always cost more than a steam engine (without a boiler), because the materials and the workmanship are much more expensive. In considering the relative cost of oil-engined and steam winches and auxiliaries, it is necessary to bear in mind that the steam plant requires a boiler which entails considerable cost and weight. Examples of oil engine winches may be seen in Figs. 7 and 8.

Electrically driven auxiliaries have also the disadvantage of heavy first cost because a powerful oil engine coupled to a dy-

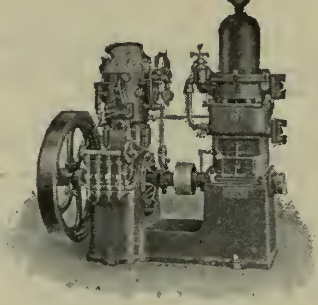


Fig. 4. Oil Engine and Plunger Pump.

namo is necessary in the engine-room, and it is difficult to get one to work economically with the varying demands on board ship. Where there is space, as there usually is in large vessels, it is best to divide the generating system into two or three units, so that one or more can be in use as required, the aim being to keep each generator as near full load as possible. This is much more economical in operating costs than having one big oil engine generator running at half or less load for most of its time, and is a safeguard against total breakdown of the electrical system in the event of mishap to engine or dynamo.

All the machines require electric motors which add to the expense besides the usual transmission losses. In the case of large vessels it is assumed that the comparatively large air compressors would not be driven by electric current, but by oil engines direct-coupled, as is the case in many large motor ships.

In some large motor vessels where auxiliary boilers are fitted, a steam driven air compressor is installed. This is very useful as a stand-by set which is used in the event of starting air being lost whilst in port, when the auxiliary boiler is generally in use for winches, etc. At the same time, it is very convenient for turning the main engines.

The advantages of powering by electricity are its great convenience and flexibility and the fact that practically all the machinery is under the eye of the engineer in the engine-room.

Economy lies in the direction of the oil engine only where it is practically possible. Dividing oil-engined vessels into three sizes, the following would appear to be the most satisfactory layout for the auxiliaries:—

(1) Vessels up to 400 tons D.W. to have hand steering gear, oil engine winches, the forward one to work the windlass by a messenger chain, a combination set in the engine-room to operate the pumps, the air compressor, and if necessary, a belt driven dynamo for lighting.

(2) For vessels from 400 to 1,500 tons D.W., some form of mechanical steering gear is necessary. Probably the most economical would be an electric generating station for the steering gear, winches, capstan and windlass, although in a number of vessels, hand steering gear is accepted, even up to seven or 800 tons D.W., in which case, the previous suggestion would probably be adopted.

(3) Vessels above 1,500 tons D.W. would no doubt have electric-hydraulic steering gear fitted direct to the rudder head, the power being supplied from an electric plant in the engine-room, the winches, capstan and windlass also being driven by electric power.

In large motor tank ships, a boiler is required for steaming the cargo tanks, in which case possibly it is advantageous to have capstans and windlass steam driven as well as a steam driven air compressor. Usually the steam is only required when the cargo is being discharged in the case of very viscous oil—such as cocoanut oil or heavy fuel oil. Therefore if the boiler is lighted up just before arriving at port of discharge, the steam is ready for capstan and windlass. The "Cochran" type of boiler is very suitable for this purpose. In a case like this, the steering gear and other auxiliaries would be electric, so as to avoid the necessity of using the boiler when at sea. In large vessels, especially those trading in northern latitudes, a boiler of some kind is necessary for heating the accommodation, as electric heating is somewhat expensive.

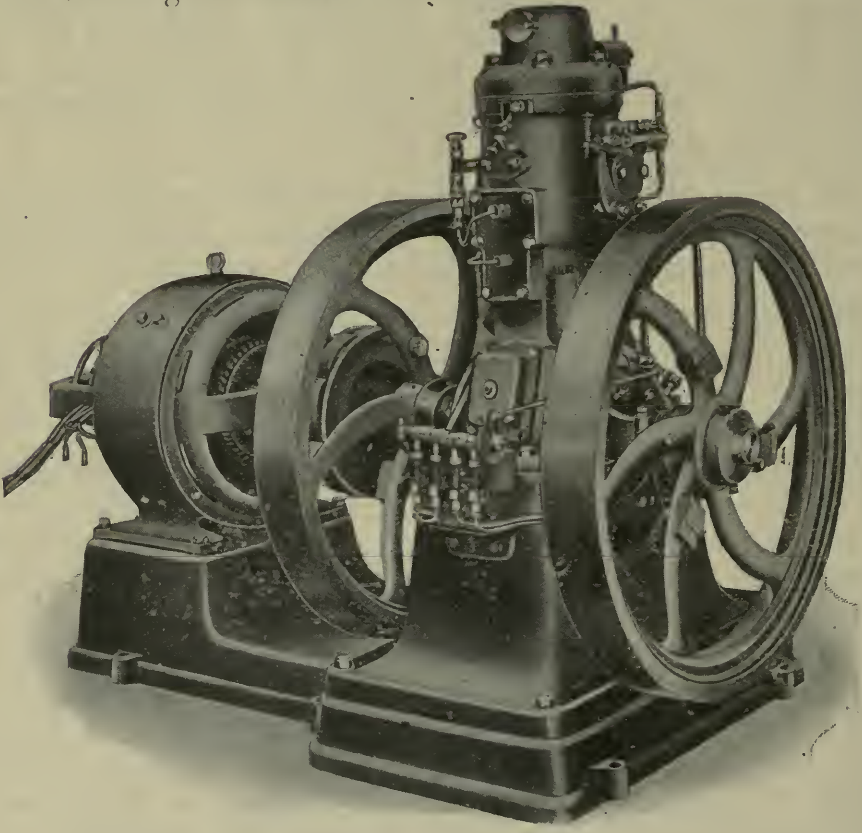


Fig. 5. Electric Lighting Set.

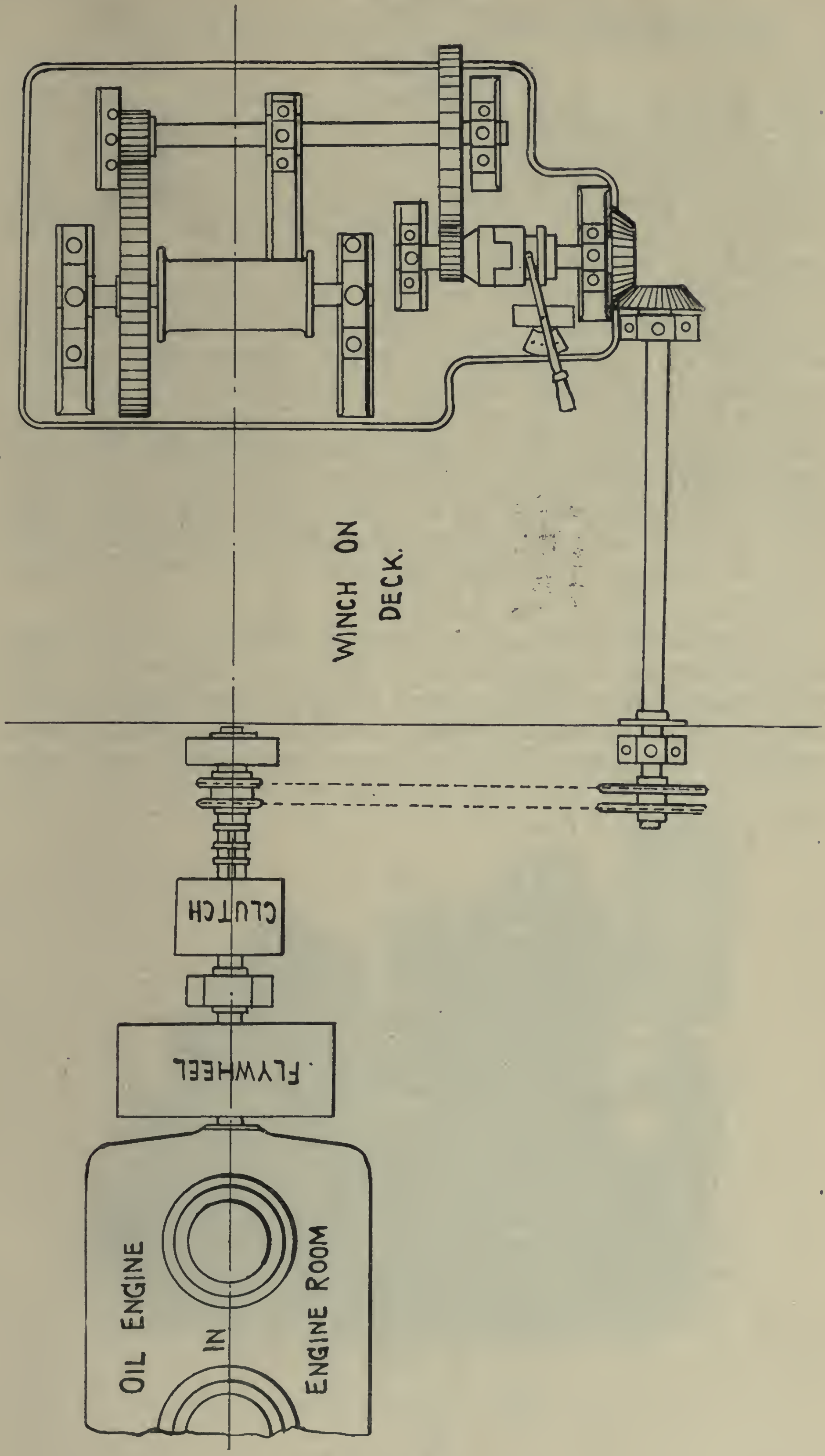


Fig. 6 Deck Winch Drive.

AUXILIARY MACHINERY OF OIL-ENGINED VESSELS.

If a boiler is fitted, it should not be in the engine-room when the main and auxiliaries are driven by oil engines, but in a separate compartment.

For oil engine driven electric lighting and electric generating sets, it is advisable to have a self-contained unit with an oil engine using similar oil to that used in the main engines.

The general question of lighting by electricity is the same for all sizes of vessels. The bilge pumps for most vessels can be driven off the main engines; as an auxiliary bilge pump is required for all classed vessels, a separate engine is necessary, which can be arranged by having an oil engine using, where possible, similar fuel to that used in the main engine, but direct coupled to the bilge pump on one end, and having a stand-by or auxiliary air compressor at the other end. This pump may then be used also for general purposes such as filling and emptying ballast tanks, washing decks and sanitary purposes. This general idea is seen in Fig. 3.

In very small vessels, an economical method of driving auxiliary pumps, air compressors, or even electric lighting sets can be arranged with a belt or chain from the main engine shaft, and instead of having tripping gear, it is better to arrange for the belt to be tightened up by a "jockey" pulley, and, when not in use, to be taken off the belt wheel on the shaft, and suspended ready for use.

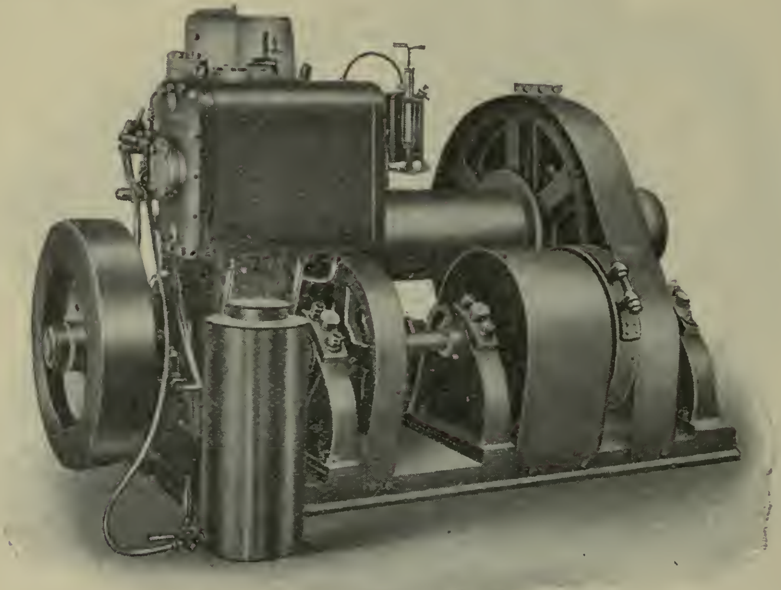


Fig. 7. Oil-Engined Winch.

AUXILIARY MACHINERY OF OIL-ENGINED VESSELS.

Fig. 6 illustrates a method of driving the cargo winch from the main engine, when the winch is placed at the after end of hatch in small coasting vessels with engines placed aft. The type of clutch is shown in detail in Fig. 9.

For small tankers, the plunger pumps for discharging cargo can be driven very economically. If the pump is fitted on the main deck, near the engine-room, the pump can be driven from the propelling engine like the winch in Fig. 6. An alternative method is to fit an auxiliary oil engine on deck in a deck house with a clutch to connect it to the cargo pump on one side and to a shaft line on the opposite side. This shaft line could have a warping drum on the extreme end outside the casing for warping the ship in and out of dock. It would also drive a bilge pump for bilge and wash deck purposes and by a belt could drive a dynamo for electric lighting, etc.

Several illustrations give an idea of some of the auxiliary engines on the market that are suitable for marine work. Fig. 3 shows a set being fitted on some oil-engined tugs for work on the River Thames. It will be seen that the air compressor is direct-coupled to the surface ignition engine, running at 600 revolutions per minute, through a friction clutch, a centrifugal

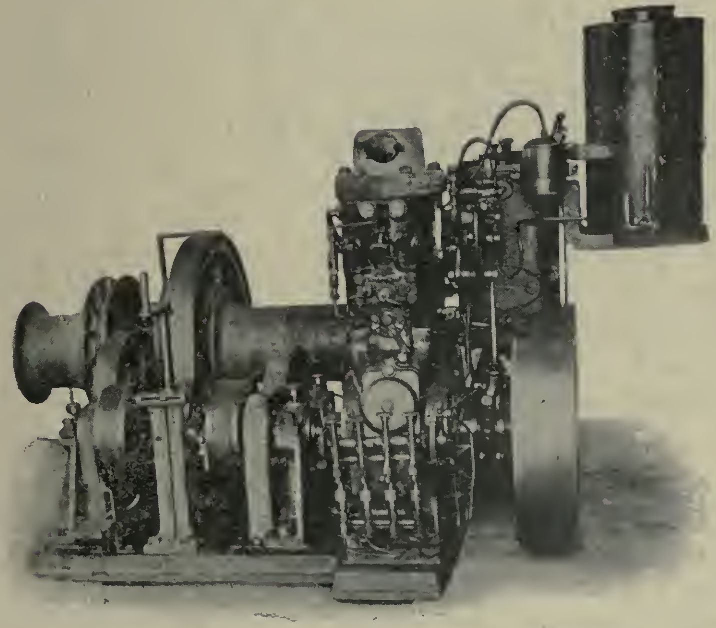


Fig. 8. Oil-Engined Winch.

pump being driven on the other side also through a friction clutch with a gear-box in addition, to bring the revolutions of the pump up to 2,000 per minute.

Another method that has frequently been adopted with vessels up to 500 tons deadweight, is to have a friction clutch to connect and disconnect the drive from the forward end of the fly-wheel, see Fig. 9. By this means, the air compressor on one side and the bilge pump on the other, can be driven when required, and can be disconnected when not in use. This method appears to be preferably to having a crank pin extension permanently connected to the fly-wheel at the forward end of the engine for driving either an air compressor or the air compressor and bilge pump.

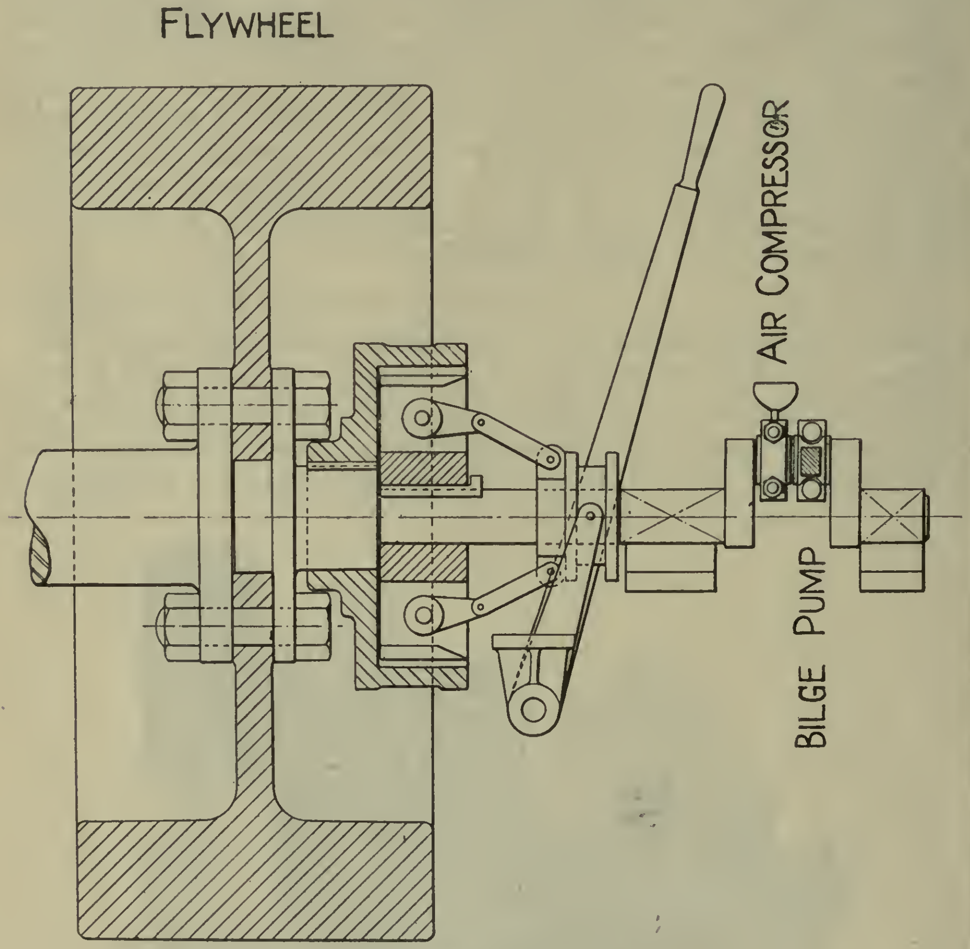


Fig. 9. Engine Room Drive.

The pump illustrated in Fig. 4 can be used also for salvage work, which is a great advantage for river and harbour vessels.

Salvage pumps that require to be self-contained units for lifting on board wrecks and sunken vessels will be handier, more portable, more convenient and more economical if fitted with an oil engine, than if fitted with the cumbersome steam boiler.

Fig. 10 shows an auxiliary set which is somewhat unusual. As will be seen, there is one engine driving a dynamo and another driving an air compressor through a friction clutch, the latter engine also driving a double plunger bilge and general service pump by means of a silent chain. For ordinary use the coupling between the dynamo and left hand engine is left unbolted, but in the event of any mishap occurring to either engine, this coupling is bolted up and the other engine is put into service for either compressor, dynamo or pump. The right hand engine is only large enough to drive the dynamo and is smaller than the other engine, but is sufficient to drive the air compressor slowly in an emergency. Of course, in the event of a broken crankshaft of the left hand engine, this could not be used.

The cost of working is the most important point when competition is keen and severe. In this respect, the oil engine will always show to advantage and in the future to even greater advantage, as the cost of the cheaper oil fuels will be reduced as the supply of oil tankers increases. On the other hand, coal will not see the same proportionate reduction. The economy in working has been shown quite recently by a 20 b.h.p. oil engine which was doing the work of a steam engine at one-third the cost. The Author knows of many cases where the oil engine is to-day working at one half the cost of steam machinery.

A method adopted by the Admiralty on several of their oil tankers in use during the war, fitted with twin sets of 320 b.h.p. Bolinder four-cylinder engines for propelling purposes, was briefly as follows:—

At the forward end of each main engine, a length of shafting was connected through a friction clutch to a "Hayward-Tyler" two-throw reciprocating pump, this pump having single reduction gearing to reduce the revolutions to enable the pump to deal with the thick fuel oil. When these pumps were in action the after clutch (for the propeller) was of course disengaged.

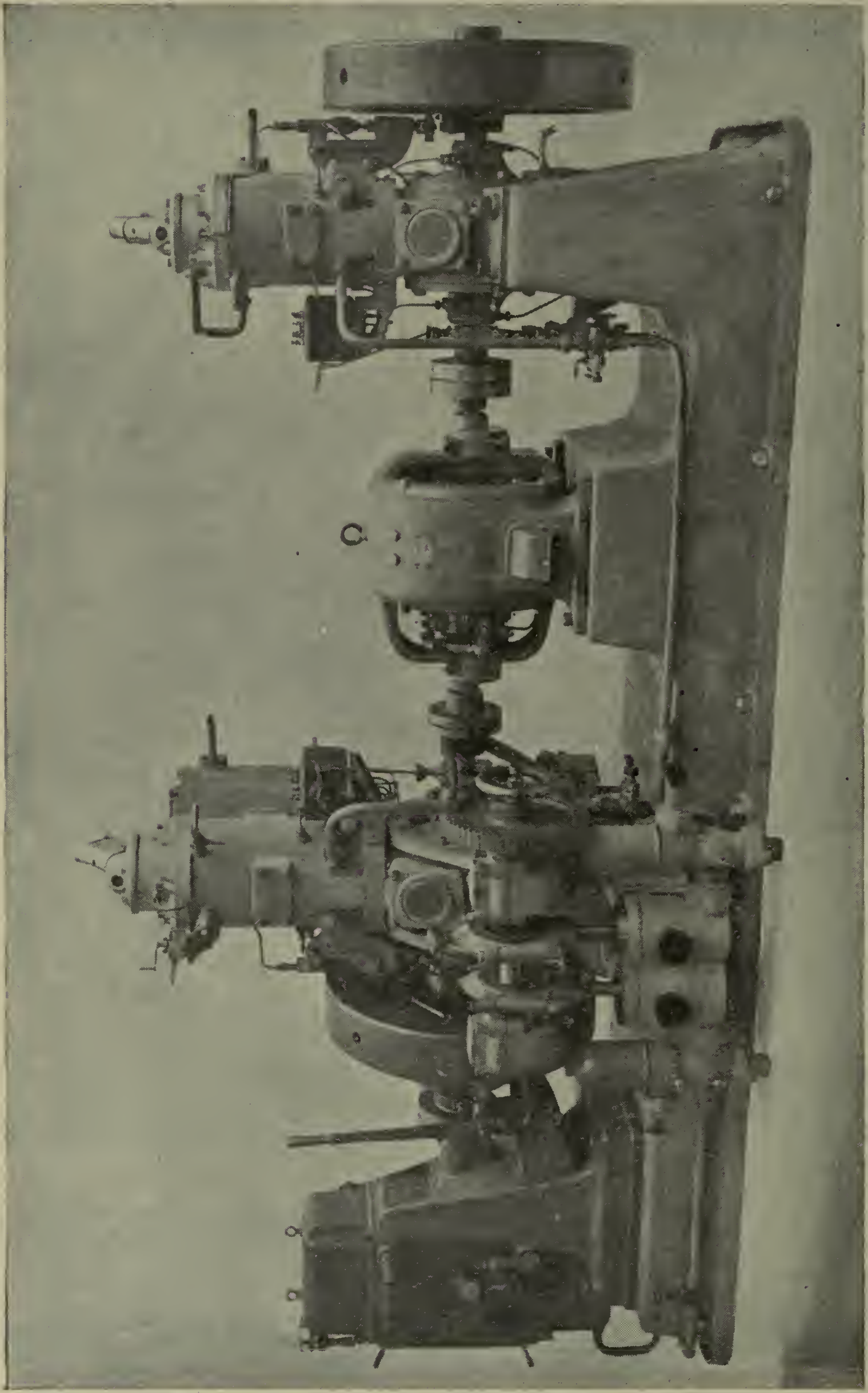


Fig. 10, Salvage Pump and Oil Engine.

Only two cylinders of one engine were required to drive the pumps. To ensure all the bulbs being kept at a sufficiently high temperature to enable the vessel to get under way instantly the load was changed occasionally from one pair of cylinders to the other pair. This method worked very successfully.

An interesting and simple pumping arrangement fitted on a 300 ton oil tanker fitted with twin 120 b.h.p. two cylinder Bolinder engine is as follows:—At the forward end of the port engine crankshaft a sprocket wheel is fitted which drives, by means of a roller chain, a horizontal ram pump placed on the starboard side of the deck, thus giving the chain a diagonal lead. A "jockey" wheel is also fitted on the slack side of the chain to ensure alignment. When at sea the roller chain is removed, thus dispensing with the necessity of a clutch. This drive has been in use nearly eight years, and has given no trouble whatever.

The Chairman:- At the start, and in no small way in a large part of the paper, one might consider that the Author was backing up the steam engine. He invariably criticises the work of the oil engines, and not at all times in their favour, bearing in mind that the steam engine has, as he states in the first paragraph, "a greater flexibility." Well, we are quite sure of one thing, and that is, that the old-fashioned steam winches when in use for discharging cargo do their work admirably. As we all know they get very severe handling; indeed, what machine gets a more severe handling than a steam winch? It is backing and filling alternately, and if you are discharging into lighters with at all a rough sea it has to perform this very quickly, and much as one, speaking as an engineer, regrets that that is so, because of the repair work it entails, still one must readily admit that it cannot be helped. With regard to the flexibility of oil-driven winches, the question is—Will they do the same? And will they be so handy? Does the oil-driven winch work continuously or has it to be started at each operation? If so, that means the efficiency of a clutch coming in, and the efficiency of reversing gears. You do not have the trouble with a steam winch. You have it acting at once. I do not think we should question whether one or the other for the moment, is directly the more efficient, but we should take the whole matter over a period of time, and see whether in the steam winch we have not something more valuable, and something more suitable to the men that usually handle these than what the oil-engined separately driven winch would give us. A little further down in the paper is a mention of winches being driven from a central part of the ship, and if you look at the diagram you will notice that there is a vast amount of bevel gearing. Well, we are all pretty well assured that driving through bevel gears, or any other gearing for that matter, entails a loss. What is the loss going to be in driving these four winches. I now refer to Fig. 2. Again, that central drive would have to be going continuously. You cannot say that all of these four winches are going to be hoisting, lowering, and stopping at the same moment. Further, that engine will have to be sufficiently large to drive the four at any moment it is called upon, so, in an average case, it might be four times as big as necessary because of only one which might be working. Do not run away with the idea that oil engines are not coming. I believe they will have their way in many cases, but at the same time I think that for auxiliary machinery there is still a big opening for the donkey boiler and the steam driven winch and windlass. In each of these cases no doubt the general lay-out has an advantage, but I query very much if it has that great advantage that we see in some steam auxiliaries. Then as regards the so-called inefficiency of the steam winch. Many of us perhaps remember that at the last Machinery Exhibition at Olympia there was an enclosed winch shown there: it struck me, and no doubt many others, why nobody had thought of doing such a thing years ago. It was a closed-in winch, absolutely working in oil. And yet for years and years we have allowed winches to work under the worst conditions you could expect.

A Member: It seems to me, taking the paper as a whole, that the author is not sure of himself. If you are speaking of steam driven vessels I can quite well see that you might introduce some of these fads—I cannot but call some of them fads; but if we are speaking of oil-driven ships, let us stick to the oil-driven conditions right throughout the ship. Improve that. If I could not get steam I would be inclined to have hydraulic power, provided we do not get into freezing Northern latitudes. To try and make a workshop or a factory of a ship by introducing line shafting—well, you want cargo space. You cannot have all that shafting below, so you must have it on deck standards. It is going back to a worse condition than we had before we started. Coming to the first paragraph again about steam winches, I would like to see an oil-driven winch tackling the conditions that used to prevail on the Suez Canal at one time. I recall one instance in particular of a 5,000 ton deadweight ship drawing 24 feet of water, and the Canal was supposed to have a depth of 26 feet. That ship was smelling the ground over every inch, so that she went from one side of the Canal to the other. Let us see what an oil-driven winch would do to draw such a ship off the banks when she got athwart the banks. Nothing but a steam winch or a hydraulically driven machine, or operated machine, is fit for such work. We must not forget that when putting a warping strain on to a ship to pull her off, that even your steam winch may be brought up all standing. The steam is still in the cylinder of the winch ready for the least "creep" that shows that the ship is giving way. An oil-driven winch would be standing quite still all the time.

About winches—the slowest machine I have ever seen turn round was a hydraulic winch, and it took two minutes to make one revolution. You can reduce it down to that speed and be effective. I would recommend to the author's attention the hydraulically operated or hydraulically controlled gear. There is no gainsaying the beautiful steering gear that has now been introduced. It is done electrically first, and hydraulically in the last operation. Of course the question would come in about an accumulator. There is so much heat being lost with the internal combustion engine which would provide no doubt a steam load for such an hydraulic accumulator, and also to provide hot-water for the whole ship's company, and for the ship's use. In regard to the use of electrically-driven winches, I may say that there are a good many electrically-driven winches in use as transporters in warehouses in London. They work very efficiently, and no one could desire a better tool electrically-driven. They should certainly commend themselves for working over hatches. They can travel in a fore and aft direction anywhere you like, but not sideways.

The Chairman: Reversible?

The Member: No. They are continuous running. The reversing is done by friction gear.

A Member: I have been encouraged to rise this evening, because of the remarks the chairman made, first, with regard to the old-fashioned steam winch for which he has shown a predeliction, and also the criticism which I think was well judged of the design shown in figure two, with its shafting and gears, which has also been fallen foul of by the previous speaker. Some years ago I prepared for the Institution of Naval Architects a paper dealing with the question of auxiliaries at sea. At that time marine engineers had not had anything like as much experience as they now have obtained in the use of electricity on board ships fitted with internal combustion engine propelling machinery. We are, all of us, to-day familiar with the Hele-Shaw system of electric hydraulic steering, and we are familiar with the electric winches which probably have forged their way to the front as much by the persistent use of them in ships engined by Messrs. Burmeister and Wain, as, perhaps, by the efforts of any other single firm, and marine engineers to-day who sail in Diesel-engined ships are fairly familiar with electrically-driven deck auxiliaries. At the time that I read the paper—it was at the Spring Meeting of the Institution, in 1913—I presented to the Institution the results of a series of experiments I had made from time to time during a period of two years previously on the use of air for deck auxiliaries. At that time the field had not been very well tested. Electricity was just coming in, but it was viewed with a great deal of suspicion by sea-going marine engineers, who had learnt to trust the old-fashioned steam winch. Many a scornful criticism I heard in those days as to what would happen in a cargo boat with a crew of Dagoes if something went wrong on the West African coast with electrically-driven winches. But times have changed, and that criticism would not probably be so frequently urged to-day as it was then. The author, in his paper, divides the craft he is dealing with into three classes. He first takes small vessels up to 400 tons deadweight; then in class two he deals with vessels from 400 to 1,500 tons deadweight; and in his third class he includes larger vessels. I presume he has in mind the use of internal combustion engines of the hot bulb or commonly called semi-Diesel type in the smaller vessels, and in the vessels in his second class; but I take it that this Institute wants to view the situation from a broad standpoint, including the use of these oil auxiliaries for all classes of vessels. It seems to me that in the larger ships of 5,000 tons and over, where to have electricity is almost a *sine qua non*, the electric generator in the ship will either be driven direct by an auxiliary oil engine either of the hot bulb or smaller direct Diesel type, or if it is a larger oil tanker, as the Author has pointed out, where they will want steam for the holds, then the auxiliary can be steam-driven. You could have your electric lighting set driven by a steam engine and the auxiliary compressor necessary in every Diesel engine-room will be similarly driven. But leaving the oil tanker out of the question, and dealing with larger ships, it seems to me that the main auxiliary compressor in the engine room from the point of view of economy will probably be driven by an oil engine. Certainly the electric generator will be driven by an oil engine, and probably the deck winches and steering gear will be electrically driven. But when you come to the second class of vessels, of from 400 to 1,500 tons dead-weight—for the moment I am leaving out of the question the first class of up to 400 tons, because the author indicates that hand steering would be satisfactory—there is a field for the recommendation of the use of compressed air. Our chairman has remarked upon, and the previous speaker has confirmed his experience in the Suez Canal, the flexibility and reliability of steam winches. The steam winch stops when the strain is too great. The steam is just behind the valve ready when the strain gives, and the winch goes on. Steam being a gas, or nearly so, and air also being a gas, it is obvious to you all that what I have said of steam, is equally true of air. At the same time, all who have used air in more or less large quantities realise that while it is a very handy and useful servant it is a somewhat inefficient servant for the reason that only a portion of the energy that is put into the air to compress it is given out in the prime mover using the air. Take pneumatic tools for instance. We can disregard the small percentage of the energy used, because the great charm is that we get a tremendous amount of work done in a short time, as compared with hand work. When you come to larger machines, whether it be deck winches or whether it be pneumatic steering, it is obvious that marine engineers will want to consider carefully the question of overall efficiency. In 1912, I think it was, I began some trials in a line of coasting vessels from Liverpool, which came round the coast, and several of which came into Ipswich, where I reside. The vessels were fitted with 5-ton winches, by Wilsons, of Liverpool. I was struck with the very small average weight of the bales or articles lifted. Thinking this might be peculiar to goods coming to Ipswich I had them checked when the vessel was unloading at Liverpool. It was a somewhat different type of cargo there. We found that the weight of the lift varied from 6 to 12 cwts. Now by putting a pressure guage in place of the drain cock on the valve chest of the winch, and by wire-drawing the stop valve on the winch, we were able to ascertain what was the pressure at which the winch operated, and to our surprise and pleasure, in view of the results we were looking for, we found that from 10 to 15 lbs. pressure on the winch gave every time all the pressure we wanted from the winch at full speed, to lift the articles out. At Liverpool we found the pressure never reached 20 lbs., although the weight of the articles averaged 12 cwts. I think it will be within recollection of most of the members here that the percentage of heavy lifts that you really get in dealing with cargoes is very small. This leads me to the point that compressed air applied to winches need not be applied at 100 lbs. pressure. And the reason is that it is wire-drawn before it gets to the winch because you never have the valve fully open. Dealing with steam, marine engineers know perfectly well that the principal amount of heat required is to overcome latent heat and produce steam from water. They know it takes only 2-3 per cent. extra heat to give 100 lbs. pressure. In my experiments dealing with air I have been struck with the fact that many marine men and others are rather fond of drawing a false analogy between steam and air. They are perfectly right in using 100 lbs. with steam. They are right because it costs practically no more than 10 lbs. pressure, the extra heat required being negligible, having overcome the latent heat. When dealing with air, every extra pound pressure you demand over the pressure actually required means using so much extra horse-power that has to be supplied by the prime-mover, whether a steam engine, oil engine, or electric motor, to the air compressor supplying the air. We found by these experiments that about 1-7th of the power was required to supply the requisite air at 20 lbs. pressure for winches, as compared with what would be required for a 100 lb. steam winch. That brings me to the line I am trying to follow, viz., the use of compressed air at low pressure direct from a simple air compressor in the engine-room taken to winches at 20 lb. pressure. It need not be more. The losses due to leakage are very much less. Only one pipe is needed from the air compressor to the winch, because there is no return or exhaust pipe needed at all. If suitable arrangements are made by gearing then on the few occasions that a heavy lift, a full lift is required, you can still use 20 lbs. and lift that load more slowly. A load representing the maximum lift is usually of a bulky shape, and cannot be lifted at the full speed of the winch because you have to be careful that it will clear the coamings of the hatch, and it is handled more slowly. In regard to steering, we investigated the question as to how far an oil engine-driven vessel with a low pressure air compressor attached to a Bolinder or other hot bulb engine as a main auxiliary could be used, and how we could arrange for steering at sea. Obviously the ordinary steam steering gear will operate just as efficiently with air as with steam. Consequently the same gear which everyone is well acquainted with can be used in this intermediate class of vessel, and the air compressor is used in the engine-room to supply air. The quantity of air used for steering is surprisingly low. Messrs. Raylton, Dixon and Co., in 1913, made some experiments in regard to this in a 5,000 ton ship, the *Berwindvale*, during her trials and on her voyage to New York and back. On trials on the measured mile there was, of course, a considerable amount of air used, but for an extremely short space of time, while the vessel was being put hard over. But on the run home to Middlesboro' the quantity of air used to keep the vessel on her course was extremely small. The quantity measured only 20 cubic feet free air per minute, and the power required to produce that was small. Going to America and back, observations showed that in fine weather, with a skilful helmsman—and there was quite a large percentage between a careless and a skilful one—and under good conditions and in fair weather the quantity of air used did not exceed something like 35 to 40 cubic feet per minute. In heavy weather we had eight hours' observations taken and the average quantity required was something like 90 to 95 cubic feet per minute. The results of this investigation led to the installing of this type of steering gear in an all motor ship, the *Rolandseck*. She was built by Tecklinberg Co., and engined by Carels. She had a simple form of air compressor driven by levers from the main engine, the arrangement being similar to what we are accustomed to in steam ships, the pumps driven by levers. The *Rolandseck* behaved admirably until the beginning of the War, and I have not heard of her since. In the case of *Wotan*, fitted with 2,000 h.p. Carels' engines, a single screw was fitted with the same form of steering engine compressor and it was highly satisfactory. The British Government fitted the same device in H.M.S. *Servitor*. In vessels of from 400 tons to 1,500 tons, if retained, and supplied with air from a simple compressor in the engine room it seems to me a direction in which air can be used efficiently and the old forms of gear we are acquainted with can be adhered to. It seems to be much more flexible than to use oil engines on deck to operate winches and for steering. I do not urge the arrangement as better than, or as economical as, electric auxiliaries. But I do say there are many vessels of medium tonnage where simplicity is the main factor. For such vessels, steam steering gear can be used with compressed air, and the winches operated with compressed air with great efficiency. The previous speaker referred to exhaust heat. That also has been tried and is quite efficient. The exhaust heat from the main internal combustion engine can be used to heat water. There is a great amount of heat in exhaust gases. You cannot raise steam to high pressure by exhaust heat, but there is any amount of heat to heat water. So the accommodation could be heated with hot water, just as central heating is done in America. The waste heat from the internal combustion engine can be, and has been, used in that way. We have had some interesting experiments during the past year or so in connection with trawlers, where the steam trawler has been replaced by a trawler with internal combustion engines; and the problem has been the problem of getting the trawl. The custom is to have a boiler specially installed to operate the steam winch for getting the trawl. In its place we have offered the fisherman a simple compressor driven direct from the internal combustion engine which drives the boat, and compressed air is led direct to the capstan for the trawl, which is precisely the same as a steam capstan. The boiler is done away with and there is no need to carry oil or coal for the boiler. The capstan operates perfectly well with that supply of compressed air. Also in smaller boats for line haulage we are installing small compressors which operate the winch which does the line haulage. It seemed to me that, at this stage in the consideration of what is a highly important matter for marine engineers, namely, what is the best and simplest form of auxiliaries for the vessels they are responsible for, the use of compressed air has very great claims, and I thought it was worth while to lay the points which we have gained from experiments before this Institute.

A Member: Regarding the deck machinery for Diesel-engined ships I quite agree with the author, as I think steam winches out of place altogether. We all know the steam winch is not so efficient as it might be for several reasons, viz.: it receives very bad usage and in many cases lacks the opportunity of being overhauled as often as one would like, also the condensation which takes place in the pipes exposed to all weathers is a very great loss in efficiency. In my opinion, the best deck machinery for all classes of Diesel-engined ships, irrespective of size (also large passenger ships), is the electric winch. These winches certainly have proved very efficient in ships that have come under my notice, and have given very little trouble. Regarding the remarks of the last speaker about air for driving winches, I should like to ask one question, viz.: He speaks of only requiring 20 lbs. pressure on the main pipe line to the deck. For say eight winches, what size pipe would be required to keep the winches running at full load and maintain the speed required? Then if air is used you have the trouble of a large air compressor always running in port, and as far as my experience goes they are very troublesome. As for having oil engines on deck for driving winches, through shafts fitted with pinions and bevel wheels and all sorts of clutch gear to enable you to stop any winch required without stopping the engine, I am afraid the gear would give a lot of trouble and be always breaking down just at the period when engineers look for an easy time in a home port after being away for about six months.

A Member: It is rather presuming on the courtesy of the author to allow me to speak again, but the answer to the question I am asked is this, that I have used air with vessels with eight winches and we used the same pipes as they had for steam. I can quite understand the speaker's question, because he realises that air at 20 lbs. pressure takes up much more room per lb. weight than at 100 lbs. pressure. But I think he has overlooked this, that the actual steam pressure used in the cylinder of the winch is so low for ordinary lifts. You may put in a winch of five tons, but you very rarely have to lift 5-ton lifts continually. If the deck pipes are put in of the right size to carry 100 lbs. pressure for the full load the winch will lift, it follows that when you are lifting 12-15 cwts. a much less quantity of steam will be required, and therefore to get that smaller quantity or weight you wire-draw the steam. Similarly, if you supply air at a wire-drawn pressure of 20 lbs., I am sure my enquirer will agree that the pipes of the same size are quite satisfactory. He says compressors give trouble. Well, I can only say there are compressors and compressors.

A Member: In regard to this air compressing plant that has been spoken of. About 12 or 14 years ago a gentleman designed a plant in the North of England to move locomotives in the workshop, and these locomotives were of the Atlantic type, 120 tons to 140 tons deadweight. They had 80 lbs. compressor pressure. They simply took the pipe from the wheel valve on the pipe line on wall of workshop and put it into the boiler, and turned on the valve and the thing moved. She kept sufficient air pressure in the boiler, as receiver, for what they wanted, instead of raising steam for moving. If they could get air at 80 lbs. per square inch to move a heavy, weighty thing on wheels I do not see why they should not have it for ordinary auxiliary gear on board ship. I think it is a very adaptable thing on board ship. Engineers would not have long steam pipes to take the ice out of. In regard to leakage anywhere with electric power—I have had no experience of this myself—but I should like to ask: How are you to discover it? With steam, you detect it immediately, but with air how are you to manage it? These are little points, but they are points that a practical man on board ship will have a hard tussle over.

Steam leak you may see and hear.

Electric leak you may feel by shock; rather dangerous.

Air leak you may have washed over by soap water and make it blow bubbles, for identification.

A Member: In regard to electric winches, I was with them on board one of H.M. ships, and we used them pretty consistently every day for boat hoists, and for coaling ship, taking in 800 or 900 tons, and they gave no trouble whatever. They were reversible and of variable speed. The men who managed them were just able seamen. Regarding the air compressor, we had two high powered air compressors. They compressed to 3,000 lbs., and delivered thirty cubic feet per hour at 3,000 lbs. pressure, and if sufficient sperm oil was given them they caused no trouble. Just occasionally a new fibre ring was needed.

In reply to questions: The first query came from the Chairman, and was in regard to flexibility. We will not say that oil has the same flexibility as steam. But we are always out for improvement, and one is trying to do away with the steam engine as far as one can in internal combustion engine vessels. With an oil engine ship you do not want to have boilers, but as there are motor ships which have boilers for auxiliary purposes, then you may as well make use of the steam instead of putting other machinery in. Oil engine winches and electric winches are expensive, and when you have steam on board it is as well to use it, unless there is only a small boiler as in tankers for heating out the tanks, then it is probably not worth while keeping steam up all the time at sea. At present there is such a big diversity in motor ships afloat that there are only a few alike. The ships constructed during the war as emergency vessels and immediately afterwards with any machinery that could be obtained, is perhaps the reason why steam has survived in oil-engined ships. As to overload, the steam engine would bring up all standing. You have the steam pressure there, and when the load eases off it will start up again. With the oil engine it is not so much so. On all winches of the oil engine type, with the exception of some small whipping winches, there is a friction clutch, or there should be, to let go at a pre-determined torque, so that the engine does not receive damage due to the ignition period being fixed and speed dropping, and as there is little momentum in the fly-wheel to bring the piston over compression, pre-ignition results, and eventually the engine stops. This in time would cause damage. What is needed is a properly designed clutch and there are such clutches, and I can show anybody a 25 cwt. oil winch, discharging cargo on a vessel built by my firm. It is standing its overload up to 30 cwt. without any difficulty. That winch has been designed for 25 cwts., but it will take 30 or 32 cwts. without any fear of bringing the engine up all standing. It is a new vessel and it is probable that the clutch is not yet adjusted to a nicety. When it has been in use some time the engineer will be able to have the clutch so adjusted as to let go at about 20% overload, which a good engine should stand.

In regard to the suggested loss through the use of bevel gearing. The arrangement shown in Fig. 2 is not intended to be an ideal one. You can with one engine work by this method four winches, and this is cheaper than having four separate engines for the four winches. The shafts would not cause the trouble that would be thought at first sight. The universal couplings, two of which are fitted to each shaft, will give a certain amount to any inequality, or any slight movement in the vessel. The amount of power lost in bevels is not much, provided they are machine cut, and properly supported with suitable provision for the thrust. You can get a decently cut bevel to-day, and not very expensive, that will give between 80-90% efficiency.

Some years ago, we installed in a vessel, an oil engine driving two winches by this system of gearing, and the results were very satisfactory. The power for driving four winches is not necessarily four times that required for one winch. Our experience is that for four 25 cwt. winches, 20 h.p. would be quite sufficient, as it is very exceptional that the four winches would be lifting a full load simultaneously. Enclosed gearing winches are undoubtedly very fine things, if you can get shipowners to pay for them. But shipowners want to get such things as winches as cheaply as possible, provided they do their work, that is one of the reasons why enclosed gearing has not been adopted more than it has. Obviously it is better than open gearing.

Many of the electric winches I have seen have enclosed gearing and give excellent results. The Author does not blame the inefficiency of the steam winch, but rather seeks a satisfactory method of obtaining greater economy and efficiency by utilising up-to-date means of obtaining the desired end.

A question was raised as to heating the ship with the exhaust gases. That is a very good idea, provided care is taken that back pressure is not put on to the engine. A good deal of the efficiency of the hot bulb engine and of many Diesel engines, especially the two-stroke cycle type, is dependant upon the exhaust gas escaping rapidly and quickly, so as not to put any back pressure on the piston, and not to adversely affect scavenging. If a heating apparatus can be fitted in the accommodation that does not put any back pressure to speak of on the engine, it would be a good scheme and would do away with electric or steam heating.

The question of compressed air for operating winches, steering gear, etc., opens up all sorts of possibilities on board ship. The chief trouble is that there is a good deal of prejudice against compressed air. By many of the old-fashioned engineers, the air compressor has been looked upon as a nuisance with the result that many of them have not taken the trouble to learn its peculiarities. My own particular impression is that when you only have low or moderate pressures to deal with, one need not worry about any trouble with the air compressor if ordinary care is taken, it is when you get to three or four stages and consequently high pressures, as used in the Navy for charging torpedoes that one may expect difficulties, and considerable care is then necessary to prevent trouble. The older compressors of course gave trouble, where too great a pressure was raised in one stage, resulting in considerable heat and burning of the valves. I know from my personal experience in the Navy, that a certain amount of trouble may be expected with the high pressure compressors, but a moderate pressure machine will almost look after itself.

We have fitted up a large number of air compressors and we have not had any trouble worth speaking of with the moderate pressure type. An instance as to the reliability of compressed air may be taken from the case of the ordinary railway train, where the Westinghouse brake is one of the most reliable pieces of mechanism you can have. From my own railway experience, I can say that I know that one of the things that seldom gives trouble is the Westinghouse brake, the working pressure of which is between 70 and 90 lbs. per sq. inch. The pressure has to be maintained in the train pipe or the train would automatically come to a stand-still and we all know that stoppages due to this cause are very small indeed.

I am sure those engineers who have the interests of the internal combustion engines at heart, must be very interested to know that compressed air could be satisfactorily used in ordinary steam steering gear without difficulty; this has always been one of the troubles with the moderate powered vessel, where the expense involved by fitting electrical-hydraulic steering gear could not be considered for one moment, and I certainly think that compressed air will go a long way towards solving the difficulty.

We have done a good deal of work in connection with tugs on the Thames, and one difficulty has been in obtaining suitable power steering gears for the job.

As you probably all know, most of the larger steam tugs on the Thames have steam steering gear, and for the class of work on this river, it is necessary to have something that can move a fair size rudder rapidly to enable the tug to be quickly handled.

I think that if we obtain a satisfactory power driven steering gear, one of the greatest difficulties of the oil-engined tug—particularly for river work—will be overcome.

It would also be of very great convenience to have a low pressure compressed air system on moderate size coasting vessels, as it would then be available—in addition to the steering gear—for winches, windlass and capstan, and avoid the expense of electrical gear. The only other trouble might be that the average tug driver or coasting engineer might not like the air compressor, and think it something of a bag of bricks, but he could be educated, particularly if a low pressure type were used.

Another point regarding compressed air which one speaker mentioned was leakage; one cannot see the leakage as we can with steam, but I do not think that need cause any difficulty in an ordinary vessel, provided the pipes and points are properly made to stand the pressure. A lot of people put in a compressed air pipe system but they do not know a great deal about it, with the result that the fittings and flanges are frequently too light. It is of course the same with steam, as we can have a bad steam pipe system as well as a good one, and the same may be said for compressed air.

I trust that in the future, we may see more vessels fitted with this means of transmitting power to the auxiliaries in different parts of the ship and that some reliable data may be obtained as to the cost of operating and maintenance.

A member spoke about electrical auxiliaries being the best generally for vessels. For medium sized and large ships electric gear is very good, where you carry an electrician, as one ought to, in all large ships. So none need be frightened of losing a certain amount of the current.

Referring to remarks on the subject of leakage, I do not think electric current leakage need trouble anyone very much, if the cables and switch gear are of high class quality suitable for marine purposes. In the Navy and large passenger vessels, breakdowns seldom occur. To-day there are many good types of watertight fittings. When we come to the smaller vessel, it is out of the question in ordinary cases to instal electric steering gear or winches, because you do not carry the men on board ship who could properly deal with it. Occasionally you get a man who is a good engine driver, and also a good electrician, but not as a rule. So I think electric auxiliaries will not come in for the smaller vessels for a good many years to come, because a lot of people are afraid of it. A member referred to steam piping, from which he had to take out the ice. That is a very tedious job and would not occur with either oil engine or electric winches. I might add, that with oil engines on deck and oil engine winches, you have to watch that you do not have frozen jackets, resulting in burst cylinders. It is not a very difficult job to drain off the water as there is always a drain cock, but it is a point that may be forgotten.

Referring to the remarks with regard to electric boat hoists in the Navy. I saw many of these in cruisers and battleships, and I must say I never saw any trouble with them. The various torpedo officers who used to look after the electric gear in the Navy spoke very highly of electric hoisting gear.

SESSION 1922-23.

President : ENGINEER VICE-ADMIRAL SIR GEO. G. GOODWIN, K.C.B., LL.D.

Types of Large Marine Oil Engines.

BY DAVID R. HUTCHISON.

Read Tuesday, February 21, 1922.

THE marine heavy oil engine is now recognised as a practical proposition for the propulsion of sea-going vessels. A study of the shipbuilding returns for recent months clearly indicates that many shipowners have no longer any doubt as to the reliability of the marine oil engines, and are eager to benefit, by the low consumption of fuel oil and the saving in space, which accrues from the application of this prime mover to the propulsion of ships.

For large powers to say 6,000 S.H.P. and upwards, the geared turbine will undoubtedly remain in favour for a long time in certain quarters.

For lower powers down to say 500 S.H.P. the problem for the shipowner will soon be that of deciding what particular type of heavy oil engine to install.

Certain types of oil engines have reached what we might term the "standardised" state, by reason of the gradual perfection of design brought about by the experience gained of the running of the engines over periods of time extending to almost ten years in some instances.

Other types which have been recently adopted, have still to stand the test of time, and may from the Marine Engineers' point of view, be said to be in the experimental state.

In this paper it will be found that the author deals with one of the latter types, namely, the Scott-Still engine, in more detail than with other types, and that he inclines to refer the knowledge he has acquired of other heavy oil engines to the Scott-Still engine for the following reasons. Firstly: the Scott-Still engine is the latest development in marine propulsion, and embodies certain novel features with which it is assumed the members of this Institution will be less familiar than with the details of the older types of Diesel engines. Secondly: the author has had the special privilege of having charge of the construction and operation during extensive shop trials of an experimental marine Scott-Still engine unit, developing 400 B.H.P. in one cylinder. The first of these trials took place in May, 1920. Since then an important series of tests have been carried out, extending up to the date of this paper, under varied running conditions, from which important data have been obtained.

It is fairly certain that the marine oil engine will gradually work towards a very limited number of forms, and have the same similarity of design as have all the present day reciprocating steam engines. As engineers, we are fortunate to have lived in the days of transition in marine oil engine design and application.

Assuming then, that the development of the marine oil engine will for a considerable time be left to designs of the reciprocating type, it might prove helpful to summarise the essential features of such types under the following headings:—

A. Operating cycles and fuel economy.
B. Cylinder charging and exhausting.
C. Combustion of the fuel.
D. Cooling.
E. Framing.
F. Valve gear and manœuvring gear.

OPERATING CYCLES.

The oil engine is one variation of the general type known as internal combustion engines. The internal combustion engine is the most direct method of converting the heat energy of the fuel into work. The fundamental function of the engine is to provide an envelope in which to carry out the combustion of the fuel in the presence of the requisite quantity of oxygen.

Again the energy liberated from the fuel must be available in such a manner as to perform useful work.

The convenient method of attaining these two features of the internal combustion engine is to provide a cylinder in which a piston moves, and by its attachment to the usual slipper, connecting rod and crank, to obtain the necessary rotary motion for the turning of the ship's screw.

It is found that using air as the medium in which the heat units of the fuel are liberated, certain limitations of efficiency arise.

Having obtained complete combustion of the fuel in air, an amount of work equivalent to the heat units of the fuel burned cannot be obtained for several reasons, viz.:—

(*a*) Incompleteness of the temperature reduction of the products of combustion.

(*b*) Cooling of the products by the heat radiated to, and conducted away by the envelope itself.

(*c*) Mechanical losses.

Fig. 1 shows the extent of these losses on a diagram of the usual pressure-volume kind. Such a diagram as enclosed by J.K.L.M is a practical impossibility if assumed to represent 100% combustion efficiency, but it is drawn to show the magnitude of the losses in terms of mean effective pressure in pounds per square inch of piston area.

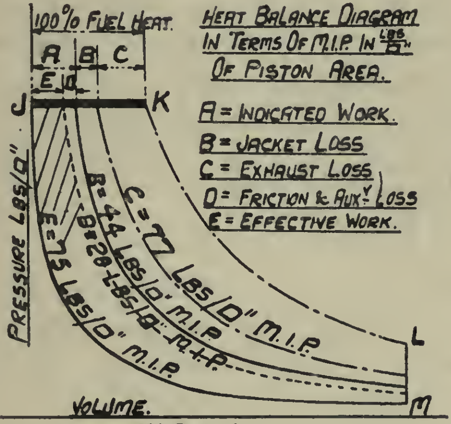


FIG. 1

It is found by experiment that the extent of the losses varies with the size and piston speed of the engine. The work obtained direct from the combustion products to the piston rod, does not vary so much as the division of the losses between cooling and exhaust.

The greatest gain in overall efficiency may be looked for in the recovery of some portion of the losses B and C, as against possible perfections of the process of combustion itself, which at the present state of knowledge, appear to have reached a somewhat flat portion of a combustion efficiency curve.

The operation of an internal combustion engine involves four distinct processes as follows:—

(*a*) Enclosure of a charge of fresh air.

(*b*) Complete combustion of the fuel in the air.

(*c*) Reduction of the temperature of the products of combustion by re-action with the moving piston.

(*d*) Discharge of the products of combustion.

It is here sufficient to state that the cycle of operations may be arranged to occupy two strokes or four strokes of piston movement, which will be referred to subsequently as two-cycle and four-cycle engines respectively.

It is found that the efficiency of the heat cycle is improved if the air charge is compressed to a fairly high pressure and temperature before combustion.

The compression pressure varies from 300 to 500 lbs. per square inch, depending on certain other circumstances, such as the degree of spontaneity of combustion desired.

The process of combustion and the charging and exhausting of the cylinder lend themselves to variation.

The process of combustion may be arranged to occur by any one of the following ways:—

(1) Combustion at constant pressure.

(2) Combustion at constant volume.

(3) Combustion at constant volume followed by a further combustion of oil at constant pressure. This method is now termed "mixed" or dual combustion.

The first method, combustion at constant pressure, is that on which all Diesel engines operate. Compression is carried up to about 500 lbs. per square inch. Oil is injected about the top dead centre and is controlled in such a manner as to give a combustion line which is practically at a uniform pressure of 500 lbs. per square inch. Cut off occurs at about 30 degs. to 40 degs. past top dead centre, and the gases expand in the cylinder until discharged at the bottom centre.

The theoretic efficiency of this cycle is about 54 to 55%, depending on the compression ratio and cut off ratio.

The following makers of engines have adopted this process:—Burmeister, Cammell Laird, Fiat, Sulzer, Werkspoor, N. B. Diesel. All these makers, as will be seen later, employ the same method of injecting the oil into the cylinder.

Constant volume combustion is not attempted as standard practice, although this method sometimes arises accidentally.

All the oil is injected at or near the top dead centre, so that the pressure rises suddenly in the cylinder. Expansion begins right from the top dead centre. For the usual compression pressure of say 500 lbs. per square inch, the maximum pressure will be considerably higher, depending on the amount of oil injected. The theoretical efficiency is in the vicinity of 59%.

The mixed combustion process has now a certain number of adherents.

In this process, a small part of the fuel charge is injected early enough to give a rise of pressure of from 100 to 200 lbs. per square inch above the compression pressure. A further part of the fuel charge is injected, such that constant pressure combustion results.

This is quite a stable process and has been adopted by such firms as Doxford, Vickers, and Scotts' in their marine oil engines.

The theoretical efficiency again depends on the compression ratio, cut off, etc., and works out about 54 to 56%.

Fig. 2 shows a typical full power card for an engine working on this process. The compression pressure is 300 lbs. per square inch and the maximum pressure 550 lbs. per square inch.

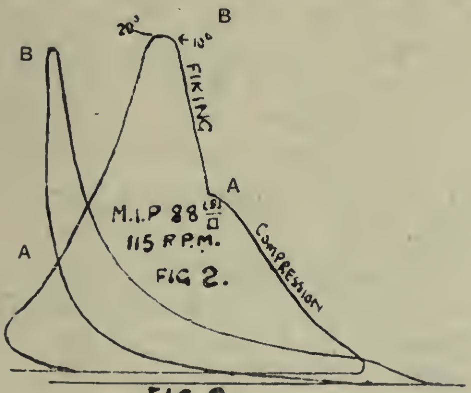


FIG. 2

The sudden rise of pressure from A to B on the PV diagram might be expected to give a certain amount of shock to the piston rod, etc. Alongside is shown a diagram which is out of phase with the crank, the top dead centre position being about the middle of the diagram. The rise from A to B is now seen to take almost 10 degs. to accomplish, and gives a slope which is not very different from that which would arise from a diagram with a compression pressure equal to the maximum pressure of Fig. 2.

The charging and exhausting of the cylinder will be dealt with in detail later; it is sufficient to note that it is regarding this part of the cycle that the fundamental difference of the two-cycle and four-cycle engine turns, and it is in the efficiency of this operation that any difference in economy of the rival cycles results.

Fig. 3 shows, to a fairly open pressure scale, the exhausting and charging process for a two and four-cycle engine.

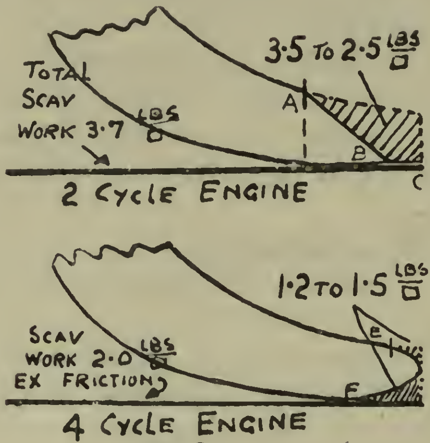


FIG. 3

Looking at the two-cycle diagram, at A the piston has uncovered ports in the liner which communicate with the atmosphere, and thereby causing the cylinder pressure to fall suddenly along a line such as AB. At B scavenge ports or valves are opened and charging of the cylinder takes place until the scavenge ports are shut on the up stroke.

The loss of M.I.P. which takes place here is about 3·5 to 2·5 lbs. per square inch.

The four-cycle diagram shows the exhaust valve opening later than the two-cycle ports, and the pressure falls along the full line E.F., resulting in a loss of M.I.P. of about 1·2 to 1·50 lbs. per square inch.

The four-cycle engine cylinder acts as its own charging pump, and the thickness of the atmospheric line represents to scale the work absorbed in this portion of the cycle, and may be assessed at about 2·0 lbs. per square inch, excluding the friction of the parts.

In the two-cycle engine, the charging air is supplied to the main cylinder from a separate source, and work has been done on it. If any of the air escapes, or loses pressure, so that when the scavenge portion of the cycle is completed, all the air pumped is not enclosed in the cylinder, then the amount lost is at the expense of more oil to be burned in the cylinder.

The ratio of the scavenge pump volume to cylinder volume is about 1·40 : 1, and the pressure should be low, say 1·5 lbs. per square inch. Referring this to the main cylinder area, the equivalent pressure would be 2·10 lbs. per square inch.

Allowing mechanical pumping efficiency as 65%, the equivalent pressure would be 3·20 lbs. per square inch.

In Fig. 3 the scavenge work is assumed to absorb 3·70 lbs. per square inch.

The losses in an oil engine are:—

(*a*) Air compressor losses.
(*b*) Scavenge or pumping losses.
(*c*) Friction (main cylinder line only) losses.
(*d*) Losses from incompleteness of expansion.

The following is suggested as a reasonable comparison of the two cycles:—

	Two Cycle.	Four Cycle.
Mean indicated pressure (complete expansion) ...	95 lbs./sq. in.	95 lbs./sq. in.
Air Compressor	9	9
Scavenge or pumping	3·7	2·0
Main cylinder line friction and small auxiliaries	8·0	10·0
Incomplete expansion	3·0	1·2
Mean brake pressure	71·3	72·8

Assuming that the oil consumed per indicated horse power hour is the same, a slight gain is shown for the four-cycle engine, in the proportion of 100 to 98.

The foregoing notes show that the difference in fuel economy of the two rival cycles is very slight, and may very readily disappear.

The two-cycle engine, from the complicated nature of the charging process, did not reach the minimum consumption so readily as the four-cycle engine. For that reason, the two-cycle engine is still at a disadvantage, but this is being rapidly overcome.

Fig. 4. The following table shows the divergence of type in relation to three important features of an oil engine.

FIG. IV.

ESSENTIAL FEATURES OF TYPES OF OIL ENGINES.

PRINCIPLE OF LOAD TRANSFER TO CRANKSHAFT.		CYLINDER CHARGING.			FUEL ATOMISATION.		
		4 Cycle.	2 Cycle.		Air Injection.	Solid Injection.	
Single Piston.	Opposed Piston.	Valves in Cyl. Cover.	Controlled Ports.	Single Port.		Constant Pressure.	Intermittant Pressure.
	Cammell Laird			Cammell Laird	Cammell Laird		
	Doxford			Doxford		Doxford	
Burmeister & Wain ...		Burmeister			Burmeister		
F.I.A.T. ...			F.I.A.T.		F.I.A.T.		
N.B. Diesel ...		N.B. Diesel			N.B. Diesel		
Sulzer ..			Sulzer		Sulzer		
Scott Still ...				Scott Still			Scott Still
Tosi		Tosi			Tosi.		
Vickers ...		Vickers				Vickers.	
Werkspoor ...		Werkspoor			Werkspoor		
Polar				Polar	Polar		

What one might call the British engines, show a preponderance as exponents of solid injection and in the adoption of the opposed piston engines.

So far we have only dealt with the cycle of events occurring within the cylinder. External to the cycle in the cylinder in most oil engines, power is not obtained and herein lies one of the principal features of the Still engine.

In the Still engine, as in all other oil engines, the products of combustion are discharged from the cylinder at a considerable temperature. 700 to 800 degs. F.

This hot gas is passed through a regenerative process, whereby the gas loses its heat to water and raises steam of an equivalent amount, at a pressure suitable to the conditions governing the use to which the steam is applied.

The heat which normally passes to the cold jacket water of the usual oil engine is, in the Still engine, a further source of steam generation at the same pressure as that raised from the exhaust gas heat.

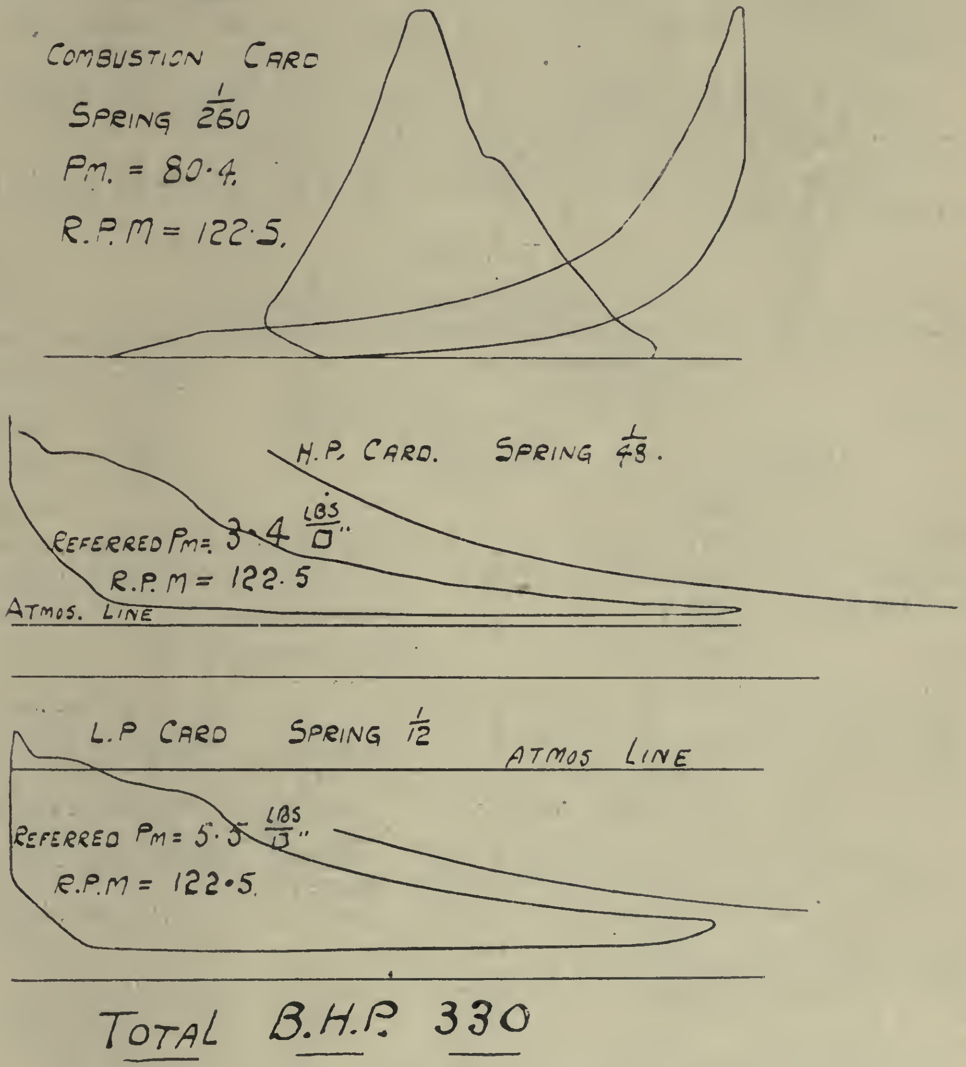


FIG. 5.

The steam pressure is roughly 120 to 150 lbs. per square inch with a corresponding temperature of 340 degs. F.

The Still engine is really cooled by the use of a "hot" jacket, the features of which will be referred to later.

The Still engine therefore turns into steam, heat which would otherwise be lost.

The steam is collected in a steam receiver and is returned to the engine and acts on the underside of the main oil engine piston.

The power obtained from the steam is approximately 10% of the oil side and is almost sufficient to overcome the friction losses of the engine, leaving the oil side free to perform useful work.

Fig. 5 shows a complete set of cards taken from the experimental single cylinder Still unit at Greenock.

Fig. 6 shows a table of fuel consumption rates of several types of engines.

FIG. VI.

FUEL CONSUMPTION RATES.

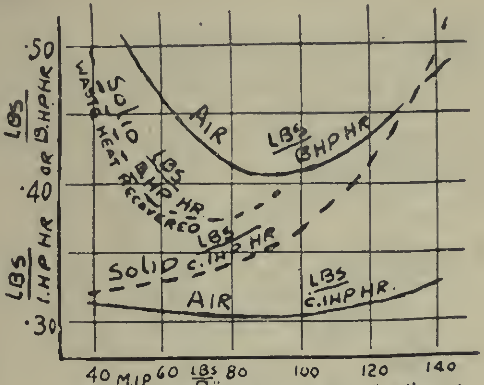
TYPE.	FUEL CONSUMPTION IN LBS. /B.H.P. HR.					KIND OF OIL USED.	TOTAL M.I.P. AT FULL POWER LBS/□″	REFERENCE. M.S. "THE MOTOR SHIP." ENG. "ENGINEERING."
	Over Load.	Full.	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{4}$			
Burmeister ...		About ·40				Diesel Oil.		M.S. Sept., 1920
(Fullagar). Cammell Laird		·391				Anglo-American		M.S. June, 1921
Doxford ...		·441				Anglo-American SP. GR. ·902	105	M.S. Mar., 1921
F.I.A.T. ...		·42						M.S. Mar., 1921
N.B. Diesel ...		·42				Anglo-Persian ·90	102	M.S. May, 1921
Scott Still ...	·398	·375	·367	·394	·476	Shale Oil ·865 SP. GR.	91·2	Eng. Sept. 2, 1921
Sulzer ...		·418	·413	·425	·50	Gas Oil ·89 SP. GR.	103·5	M.S. Feb., 1921
Tosi	·411	·42					97·	M.S. Dec., 1921
Werkspoor ...								
Vickers ...		Under ·40						

Fig. 7 shows the fuel consumption curves in lbs. per I.H.P. hour and lbs. per B.H.P. hours plotted against M.I.P. varying according to the propeller law, for solid injection and air injection engines.

The Scott-Still engine has been run since on heavy fuel oils up to the usual boiler oil ·95 specific gravity.

On Anglo-American Diesel oil, the consumption is as good as given in the table for shale oil.

The overall economy of oil engines is influenced by the consumption of lubricating oil. The consumption of lubricating oil is very much in the keeping of the engineers in charge, and a very gradual, but not reckless cutting down of the oil should be employed.



Fuel Consumption "Solid" & "Air" Injection

FIG. 7.

Cylinder Charging and Exhausting.

The four-cycle engine piston acts as its own charging and exhaust pump. An inlet and exhaust valve are provided in the cylinder cover, operated by cams suitably timed to give the maximum charge weight of air in the cylinder.

It is probably possible to enclose 90% of air at normal temperature and pressure, allowing for the contraction of the residual exhaust and momentum effects of the gas columns.

The variety in the design of this portion of the four-cycle engine is not great, usually consisting only of variations in the operating mechanisms.

One four-cycle type of engine has, however, a distinct difference in principle.

Fig. 8 shows in diagrammatic form the arrangement adopted by Tosi. The valve in direct communication with the cylinder acts on one stroke as an exhaust valve, and on the next stroke as an inlet air valve. Beyond this valve is a semi-rotary selecting valve, which can connect the first valve at the cylinder with either the exhaust or inlet passage in the cover.

The semi-rotary valve is not made a very close fit in its housing.

Assuming that the usual four-cycle engine has one inlet valve and one exhaust valve, and if again in another design it was possible to utilize both valves for inlet and exhaust alternatively, valves of half the total area could be fitted. This is where some of the gain of the Tosi arrangement is obtained. Further, the combined valve will be a cooler valve, because of its smaller size and because of the cooling effect of the charging air. The charging air is bound to pick up heat in this system on its way to the cylinder, and so reduce the charge weight somewhat, but the loss from this would be difficult to detect owing to the flatness of the fuel consumption curve for ordinary M.I.P.'s.

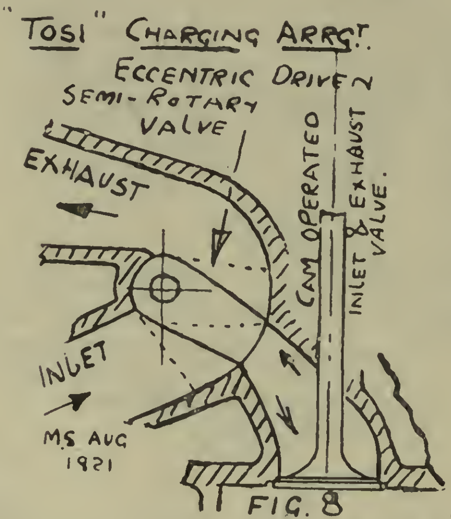

"TOSI" CHARGING ARRGT.
ECCENTRIC DRIVEN SEMI-ROTARY VALVE
EXHAUST
CAM OPERATED INLET & EXHAUST VALVE.
INLET
M.S. AUG 1921

FIG. 8

The introduction of the semi-rotary valve in the cylinder head of the Tosi engine adds to its complication, but as an offset to this, it is not placed directly over the cylinder end itself.

The charging of the two-cycle engine is a more interesting, and, one might add, more difficult problem, and it is for this reason that the two-cycle engine has been slow in general adoption.

As already stated, the charging air is pumped at a pressure to the cylinder. This pressure should be as low as possible and ranges from 1 to 3 lbs. per sq. inch in slow speed engines. Regarding the most suitable pressure to use, it is found to be that which will pass the specified quantity of air through the scavenge ports. It is not fixed by the length of sweep of the air in the cylinder.

Assuming an engine of the opposed piston type has cylinders 18in. dia. × 48in. total stroke. The scavenge period would be about 70° or ·10 seconds. For the air to move 4ft. 0in. in this time would require a velocity of 40ft. per second, which can be given by a few inches of water head.

The pressure is then dependent on the port and passage areas and these should be as large as possible.

Fig. 9 shows in diagrammatic form the scavenge principle known as single port scavenge, as adopted in the Scott-Still, Cammell Laird, Fullagar, Doxford and Neptune Polar engines. The pressure time diagram is also shown. The exhaust port closing position determines the charge weight of air which will be enclosed. The time interval between the scavenge port closing and the exhaust port closing, allows air to escape from the cylinder, unless some other means is provided.

For the same cylinder swept volume, it is possible to enclose a greater charge weight in the opposed piston type, as shown by the dotted lines in the pressure time diagram.

The whole circumference of the cylinder at each end is available for the scavenge and exhaust ports. Large areas are readily obtained and therefore low scavenge pressures.

In the opposed piston type, clean sweeping of the cylinder is obtained, as shown by the direction arrows.

Single port scavenge without other devices is only justified on the score of simplicity, as a certain sacrifice of maximum M.I.P. necessarily follows its adoption. However, in the Scott-Still engine, the recovery of heat from the exhaust by the generation of steam, permits of simplicity of the oil engine being carried to an extent which would not be desirable in ordinary single piston oil engines.

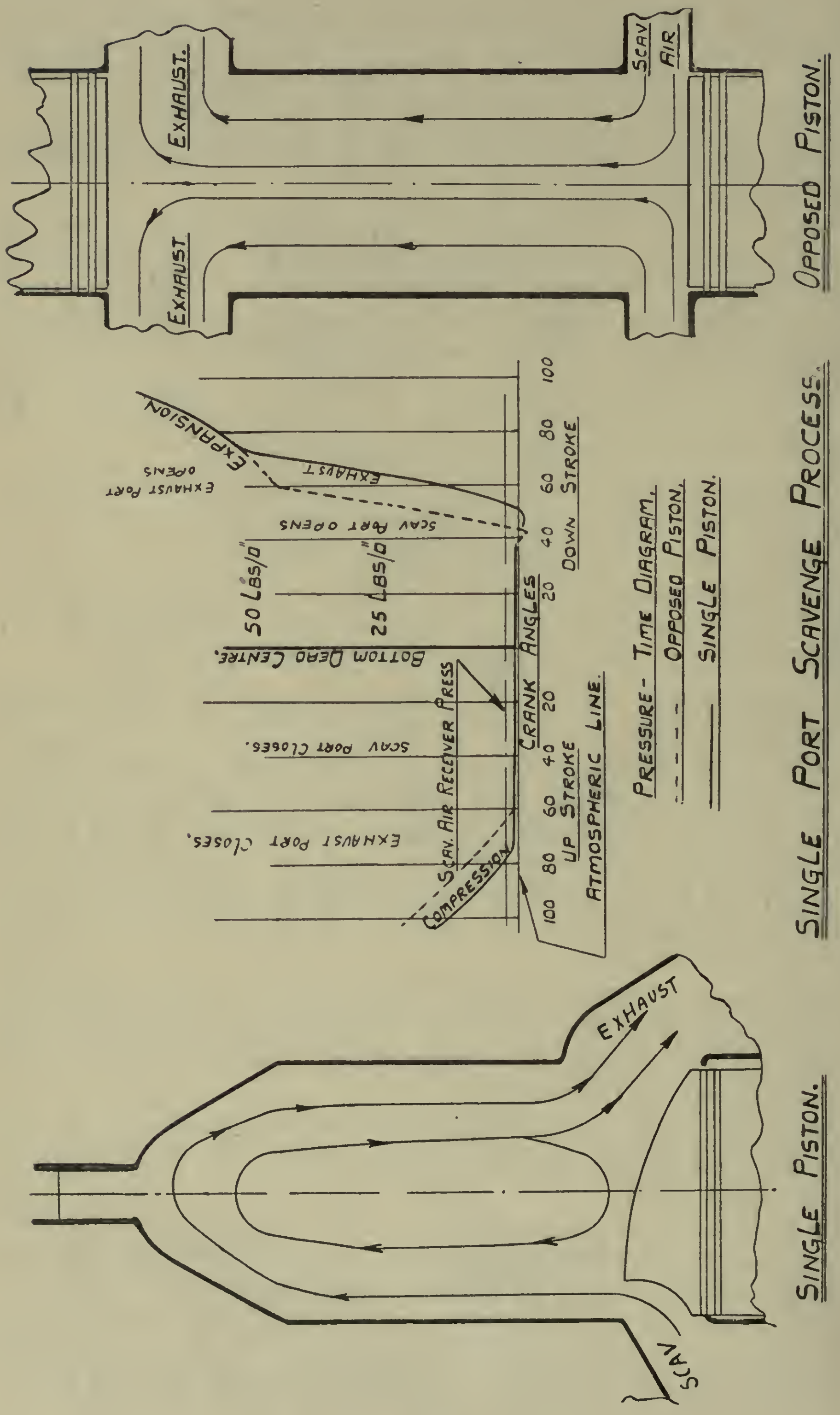
Fig. 10 shows the charging arrangement now popularly called " supercharging," and the diagrams indicate the methods adopted by Sulzer and Fiat respectively, these being the two chief exponents of the system.

The excess of air with which it will be possible to charge the cylinder, will depend on the scavenge ratio and the timing of the charging ports.

As to how far scavenge efficiency can be improved by providing direction for the incoming air, is a matter of experiment. Scavenge efficiency can be obtained in two ways:—

(a) By the clean sweeping of the exhaust in front of the incoming air.

(b) By dilution of the residual exhaust gas with fresh air.



Single Port Scavenge Process.

Fig. 9

The first method is the desirable one, provided that the air passage shapes and areas are such, that the effect is obtained without serious increase in scavenge pressures.

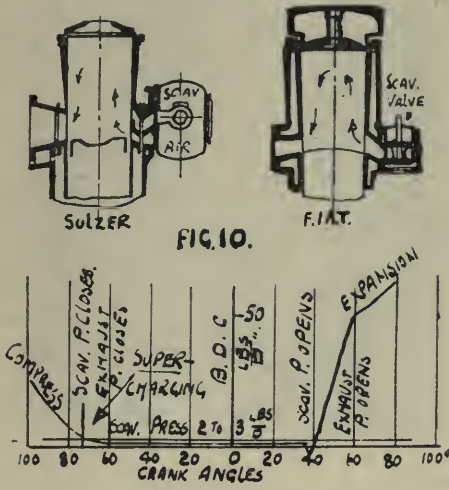


FIG. 10.

The second method arises out of imperfections of (a), but may still give results which are good measured in terms of lbs./B.H.P. hour.

The pumping of the air may be carried out in two ways.

(1) By reciprocating pumps driven by the engine.

(2) By rotary blowers driven by an electric motor or other means.

The rotary blower is now enjoying a certain popularity.

The factors governing the choice of scavenge unit, are principally two, depending on the capacity required and the general type of engine room auxiliary fitted.

The turbo blower can be driven by a variable speed motor, so that the air can be reduced to suit the engine requirements, and also on occasions, the blower speed can be increased to give overload conditions for the engine, when an increase in scavenge ratio is very desirable.

Fig. 11 shows a Reavell turbo blower which runs at 3,000 to 4,000 revs. per minute, giving 1,000 to 2,000 ft. 3/min. and is used to give air for the Scott-Still single cylinder experimental unit. Very little trouble has been experienced with the running of this machine.

The machine of this type requires to run well below its surging point to give absolute satisfaction, *i.e.*, it should be designed for a pressure adequately above the pressure required for the engine.

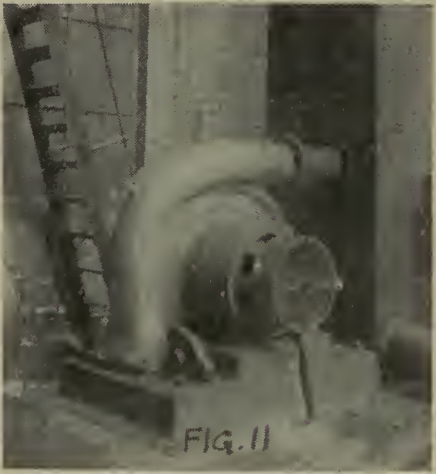


FIG. 11

Reciprocating scavenge pumps driven by the engine in either of the following ways is quite satisfactory:—

(1) Pumps driven by levers from the engine crosshead.

(2) Pumps driven at the forward end of the engine crankshaft.

(3) Pumps driven between groups of cylinders.

(4) Pumps driven in tandem with the main cylinder.

Each method has advantages and must be considered with reference to the general design of framing and ship conditions.

The Cammell Laird method of attaching the scavenge pump piston to the upper opposed piston is novel. The pump cylinder is rectangular. One wonders if the lubrication of the scavenge pump rings might not be excessive, as they would appear to be almost certain to receive more than would ordinarily be given them in another situation.

Two-cycle engines are said to reduce the number of valves on the engine. This is not quite true when one takes into account the number of valves fitted to the scavenge pump. These valves may be of automatic type, in the form of thin disc, or alternatively, positively driven piston valves may be fitted.

The automatic valve of a large total area, would seem to be the best solution of this problem, and in any case they are bound to have a much easier and happier life than the corresponding valves in the blast air compressor.

Combustion of the Fuel.

Two methods of injecting the fuel oil into the compressed air charge have definitely established themselves as suitable and reliable enough for marine engines.

The first and best known is the air injection system. In this system, fuel oil is pumped to a valve in the cylinder head by a pump which measures out the oil in relation to the power required.

At the correct moment this valve in the cylinder is lifted by a cam and the oil is blown into the cylinder by air at about twice the pressure within the cylinder, *i.e.*, about 1,000 lbs. per sq. inch. This air is provided by an air compressor which is almost invariably fitted at the extreme forward end of the crankshaft.

The second method is that called " solid " injection of which two variations are adopted.

In the first the oil is pumped up to pressures in the vicinity of 4,000 to 8,000 lbs. per sq. inch in the pipe joining the fuel valve. The fuel valve is lifted by a cam in the usual fashion, the period of the valve being such, that with the requisite size of hole to obtain the proper degree of atomisation, the pressure can be maintained. This method is usually termed " constant pressure system."

The second variation under solid injection, is that in which the pressure is applied to the valve only at the firing point, and in this case the valve is lifted by the action of this fluid pressure upon the differential spindle of the fuel valve. The valve is not cam operated. This system will be referred to as, " intermittent " pressure system.

The following may be taken as the requirements which each system ought to satisfactorily fulfil.

(1) The consumption per B.H.P. hour, low, *i.e.*, atomisation of the fuel satisfactory.

(2) Alteration of valve opening period to suit low loads.

(3) Prevention of oil entering the cylinder other than during combustion process.

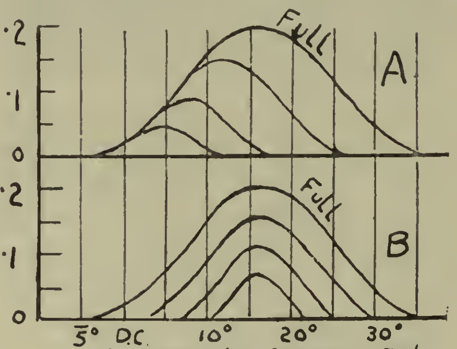
(4) Distribution of fuel to cylinders uniform.

(5) Ability to run at low M.I.P. and revolutions.

With regard to point 1, the solid injection systems will give consumptions measured in lbs./B.H.P. hour as good and better than the air injection systems up to a M.I.P. of about 100 lbs. per square inch.

Fig. 7 already referred to, shows something of the nature of the comparison of these types. So far the consumption measured as lbs./I.H.P./hour for solid injection engines, is not so good as air injection, and obviously if it could be made as good, the consumption measured as lbs./B.H.P. hr. would be very much better and no question would then arise as to which method should be adopted.

Regarding point 2, Fig. 12 shows for cam operated valves, how the fuel valve lift diagram varies according to the method employed of effecting the change. Lift diagram A is the desirable one, as the initial injection of the fuel occurs at about the same position for all loads. Diagram B shows another variation in which the period is reduced symmetrically with the middle of the cam, as would be obtained by simply altering the height of the valve operating lever fulcrum. The injection is definitely late, except at full load, and this cannot be said to be a satisfactory method of fulfilling point 2. J. L. Chaloner in the "Motor Ship," December, 1920, has gone into the question pretty fully.



MECHANICALLY OPERATED FUEL VALVE LIFT DIAGRAMS

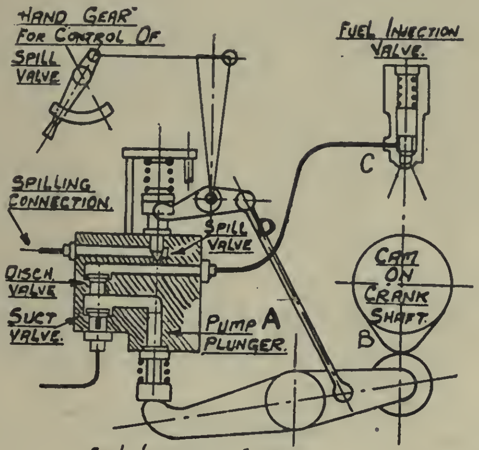
FIG. 12

In any case in order to accomplish a variation of valve period, such as A, a complication of the valve gear arises, although in some cases this motion has been obtained by the same movements as effect reversal of the valves.

The Still engine system of fuel injection lends itself to the fulfilment of point 2 in a particularly satisfactory manner.

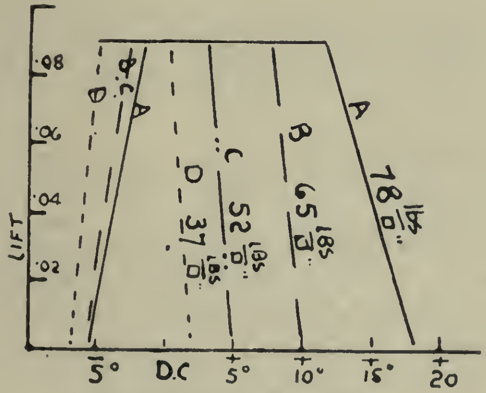
Fig. 13 shows in diagrammatic form, the complete injection system for the Still engine. The fuel pump plunger A, is operated by a cam B, keyed to the shaft at the requisite angle; a spring returns the plunger.

The plunger moving forward builds up a pressure in the fuel pipe to the valve in the cylinder. At a certain point on the travel of the pump the pressure is such as to overcome the spring loaded fuel jet C, and the valve opens suddenly and spraying commences. After a further movement of the pump plunger, and depending on the load, a spill valve D, situated in the high pressure pipe, is tripped open, the pressure falls suddenly and the fuel jet returns smartly to its seat.



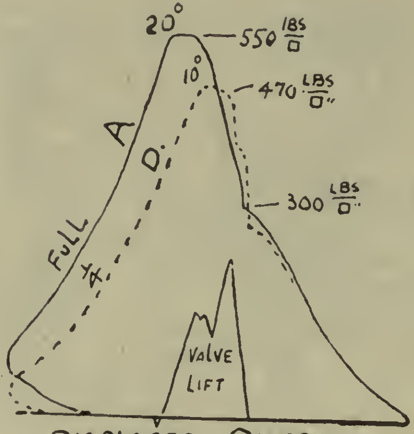
Fuel Injection System
Fig. 13

Fig. 14 shows a set of fuel valve lift diagrams taken during a varying load trial, and shows how the period varies in the correct manner.



Valve Lift Diagram
Fig. 14.

Fig. 15 shows the corresponding displaced power card for the extreme loads, together with a copy of the actual valve lift diagram obtained at full load.



DISPLACED DIAGRAM
FIG 15.

The variable period has undoubtedly been obtained in a very desirable manner, with a very simple arrangement. In this automatic type of fuel valve, the parts are so light, that it is possible to obtain shutting and opening speeds quite impracticable with cam operated valves, which have all relatively heavy parts to accelerate.

The prevention of oil getting into the cylinder except at the correct period, is a point which is fairly important. In the air injection, and constant pressure solid injection systems, any leakage at the valve face must find its way into the cylinder.

With the intermittent solid injection system used for the Still engine, this defect cannot arise, as the oil is only under pressure at the fuel valve during the period in which the valve should normally be open.

Regarding the distribution of the oil to the cylinders, two systems are in vogue.

(1) An individual pump to each cylinder, the pump measuring out the oil to the valve each stroke. This system is applied equally well to air injection and solid injection.

(2) Pumps delivering oil to a common system and subsequently distributed by throttling at a box or by arranging the fuel valve periods to give the desired effect.

The system as outlined in (1) cannot possibly give overload conditions to a cylinder without a change in the pump setting.

System (2) may, under certain circumstances, allow of excess of oil going to the cylinder.

The Still engine falls under System (1).

The ability of an oil engine to run on low M.I.P. at low speeds, depends on the temperature after compression being high enough and remaining high enough when injection commences. Air injection of the fuel causes misfiring at low speeds, due to the cooling effect of the injection air adding to that already affected by the cylinder surfaces. It may be safely stated that a solid injection engine will fire regularly at a lower speed than an air injection engine.

The Still engine in addition to this advantage of being of the solid injection type, is also hot jacketed. This still further facilitates regular running at low speeds.

For the proper atomisation, penetration and distribution of the fuel in the solid injection systems, high pressures have to be employed, varying from 5,000 lbs. per sq. inch to 9,000 lbs. per sq. inch.

This pressure sounds a bit high, but with the proper realisation of the requirements of the parts subjected to it, the system can be made extremely reliable.

Most parts are made from forged steel billets and special designs of pipe couplings must be used.

The Still engine has been run on all classes of fuel oil up to Mexican boiler oil, giving consumptions which vary practically with the calorific value.

Heating of the heavier oils is necessary to allow the oil to be handled by the pumps.

COOLING.

A very important side of the marine oil engine is that associated with the cooling of the liner, head or cover and piston crown, together with any parts, such as exhaust ports or valves, over which the hot gases career.

Ordinary oil engine practice for a long time past has been to cool the liner and head with salt water, and to cool the piston with fresh water or oil.

If fresh water were used throughout the jacket system of an engine, a higher outlet temperature could be obtained without danger from salting up or scaling, with a resulting gain in overall efficiency.

From descriptions published of new engines, one notes a tendency to introduce fresh water cooling for all purposes, and the author believes that this will ultimately become standard practice.

Oil cooling for the piston can only be justified on the argument that leakage is not harmful to the crankcase oil. This infers that given a satisfactory system of water cooling, from which leakage cannot arise to contaminate the oil, water cooling would be adopted. As to how far the present water cooling arrangements to the piston are unreliable, many of you will have some fixed notions.

In ordinary Diesel practice two methods of piston cooling are in use.

(1) Telescopic pipe method, in which a fixed pipe on the piston works into a fixed pipe in the engine framing, with or without glands, together with a third enclosing pipe to catch leakage.

(2) Link and lever method, by which the water is passed along a lever and picked off at the end by means of a flexible hose, or up the tubular elements of the link to the piston rod. The water goes up the centre of the rod and down again and away by means of the links and levers.

The telescopic pipe arrangement well and truly lined up, would appear to offer the readiest solution to this difficult problem.

The Still engine must be considered apart in relation to piston cooling. The method adopted for cooling is to direct the steam, which in any case is acting on the under side of the piston, over the surface of the piston. The actual arrangement whereby this is carried out, is to make the piston rod run into a fairly close fitting sleeve, and to drill holes up the rod to the top for a short distance there.

The incoming steam is thus made to impinge with high velocity on the centre of the piston crown. A large number of spiral ribs guide the steam outwards over the crown and down the side of the ring carrier portion of the piston barrel. The steam is thus considerably dried, and may even be superheated, with a resulting increase in steam efficiency.

The question of how best to effectively cool the liner, is one which requires more and more attention with increases in the cylinder dimensions.

That the usual practice of fitting a cast iron liner which will resist the bursting stresses and of allowing, to a certain extent, the temperature head in the liner to look after itself is not good enough in large cylinders is quite realised.

Methods have been recently adopted by several firms, whereby the liner is made comparatively thin, the necessary bursting strength being obtained by some form of re-enforcing.

Burmeister and Wain designs do not seem to have changed yet in this direction, as they still have the thick liner.

The Tosi engine has gone a step in the right direction of lowering the liner stresses, by getting more effective cooling of the liner. A form of stepped concentric channel is formed on the liner, by casting a shallow rib which is turned to a diameter to fit inside the jacket.

All the water supplied to the cylinder must pass along this channel with no doubt considerable velocity, and so giving more efficient cooling of the liner.

Provided the requisite fit is provided, the liner may be said to be reinforced by the jacket.

If salt water cooling is adopted, large inspection doors are provided in the jacket wall.

In the Still engine the cooling is obtained by means of the use of a "hot" jacket in which water is maintained at about 100 to 120 lbs. per square inch and 340 degs. F. Any heat which passes through the cylinder wall to the water, raises steam which is allowed to pass out at the top to a steam receiver. All the cylinders are connected up to a common system with a common steam receiver.

The jackets circulate by convection, no pump being used to pass the water through the jackets. Fig. 16 shows a diagram of the liner and jacket construction.

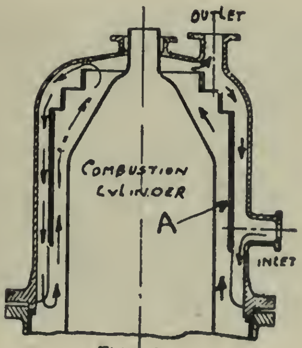


FIG. 16

The convection currents indicated by the arrows, are materially assisted by the presence of the stiffening drum A, fitted over the radial ribs cast with the liner.

As compared with ordinary Diesel cooling practice, the advantages of the Still system may be stated briefly as—

(1) Heat to cooling medium raises useful steam.

(2) Hot jacket enables low compressions to be used.

(3) Very uniform temperatures for the whole cylinder liner.

(4) Natural reinforcement of the liner from the water pressure in the jacket.

(5) Thin liner gives low temperature stress.

As the cylinder jacket is under pressure similar to that in a boiler, this part is made of cast steel.

In the Still engine a large proportion of the steam raised is obtained by temperature reduction of the exhaust gases after leaving the cylinder.

The gases are passed through a regenerator, placed close to the cylinder, and of sufficient surface to reduce the gas temperature to such a point as to allow of uncooled exhaust piping being used thereafter.

Valve Gear.

The usual method of operating the valves of an oil engine, is to provide a shaft with cams, and to have an intermediary such as a push rod and lever, or lever only, between the cam and the valve.

Together with the operation of the valves in correct phase with the crank. there is the necessity to provide reversing mechanism for the astern running in marine engines.

These operations are made to take place in a variety of ways, and a detailed description of the mechanism is without the scope of this paper, and it will be sufficient to state that ahead and astern cams are provided, with means whereby the lever is put into contact with the desired cam. Three methods of effecting this selection of the cam are in use:—

(1) Sliding the cam or the camshaft as done by Burmeister and Wain. N. B. Diesel, etc.

(2) Moving the roller or engaging a second roller as done by Tosi, Sulzer and Scott-Still, etc.

(3) Shifting the lever fulcrum as done by Werkspoor.

All these methods require that the lever be lifted clear of the cam before shifting to the other cam.

The four-cycle engine is as easily reversed as the two-cycle provided that the number of cylinders on the engine is such as to give a starting position anywhere by having the starting cam profiles overlapping in phase. More than four cylinders are required for this.

The operation of the valves of the Scott-Still engine is carried out hydraulically throughout the whole engine. The operation of the fuel valve has already been described.

The valves for the steam engine are operated as follows:—Oil is pumped to a pressure of 400 lbs. per square inch by a special pump provided for valve operating. The oil is passed to a distributor driven at the engine speed at the crankshaft, through the usual spiral gears.

The distributor consists of a plug rotating in a fixed barrel, with ports arranged in such a fashion, that oil can be supplied to adjoining pipes and released again at an interval as required by the particular valves at the cylinder to which the pipes are led. At the cylinder, the pipe is connected to an operator attached to the valve to be operated. The operator is simply a plunger which the admission of oil under pressure moves upwards, and on the release of this pressure, the valve returns under the action of oil at 400 lbs. per square inch on a differential spindle in the operator, very much after the manner of the usual spring return for cam operated valves.

The action of the valve is absolutely consistent, and as can be seen from the steam cards shown in Fig. 5, clean opening and closing of the valve takes place.

The operating mechanism is all at the crankshaft.

Fig. 17 shows the distributor for the steam valves for the single cylinder unit. The handwheels A are for alteration to the cut off.

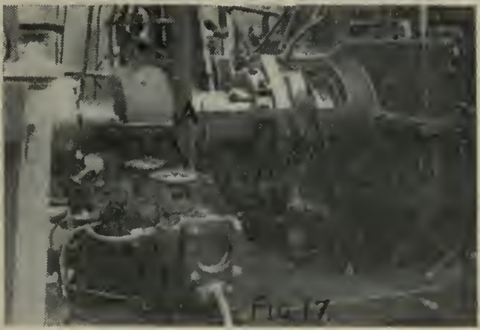


Fig. 18 shows the valve and valve operator which is contained within the steam cylinder cover.

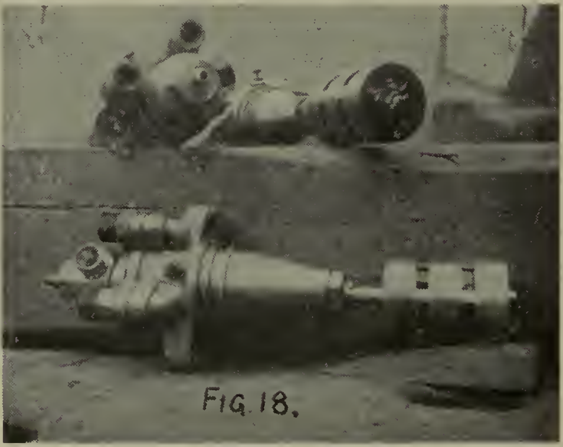


FIG. 18.

In all Diesel engines, the starting and manœuvring of the engine is carried out by compressed air.

The air may be stored at 300 lbs. per square inch as done by Burmeister and Wain, or at the maximum pressure of 1,000 lbs. per square inch, as adopted by most other makers.

The capacity of the reservoirs fitted enables a limited number of reversals to be carried out, probably about 20. Extensive manœuvring with little time between, would entail the starting up of the auxiliary air compressor.

Any difficulty of starting the usual Diesel engine arises only when quite cold, and one notes that several makers are now fitting steam heating connections to the jackets, to make this operation surer. When hot, the engine fires very often too readily, and very high pressures may result with rash handling. It is probable that the crankshaft suffers more in manœuvring, than in the prolonged running at sea.

The fact that the starting air valve on the cylinder is idle during the steady running of the engines, which may extend into weeks in the case of marine engines, causes a little uncertainty of action when suddenly called into use.

That this is no bogey, the author knows from experience, and one notices the tendency in ordinary Diesel practice to fit a protecting valve for the starting air valve, together with lubricators for the internal parts of the valve.

Again the author does not propose to enlarge on the details of starting and manœuvring except in the case of the Still engine.

The Still engine is peculiarly fortunate with its manœuvring arrangements, in that a familiar working medium, steam, is used.

As the undersides of all the main pistons are steam engines, the engine is self-starting in the usual steam engine sense.

The steam valves being always in use are not likely to give trouble on manœuvring.

There is no necessity to hurry the starting process in the Still engine, as, due to the hot engine, firing is a very certain operation.

When one considers the question of flexibility, the Still engine again shows up in a favourable light.

The engine has the usual flexibility of a solid injection oil engine, with the additional advantage of having a hot jacket for the cylinder. As it is possible to run the engine on steam alone, very low revolutions could be obtained.

It is possible to run the Scott-Still engine at M.I.P.'s of up to 40 lbs. per square inch without cooling the piston, so that if the steam pressure requires raising up fairly quickly, by shutting off the steam to the engine, the steam receiver pressure will come up very quickly, without separate firing, the engine waste heat being sufficient to do this.

Fig. 19 shows a photograph of the starting and manœuvring gear. A is the steam stop valve, B is the fuel pump hand con-

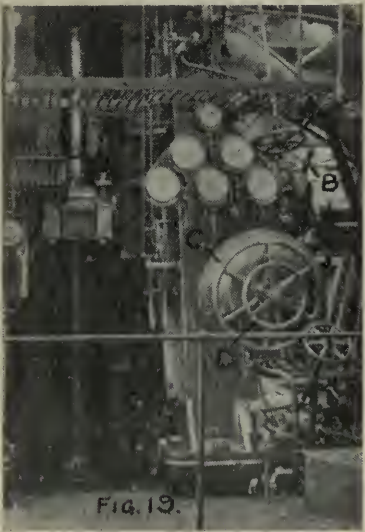


FIG. 19.

trol, but which may be engaged with a servo motor controlled by the governor, having speed regulations by the handwheel D. E is the cylinder drains.

Complete control of the engine is effected by a movement of handwheel C to the right for ahead, and to the left for astern. In each direction starting up is accomplished by turning C into three successive positions. Handwheel C is shown in stop position; a movement to the right of about 20° sets all the valves ahead. A further movement of 20° allows steam to the engine, with a cut off of nearly 100%; when the engine has accomplished a revolution or two, the wheel C is turned to its extreme position and the fuel oil is automatically cut in, at the same time the steam cut off has been reduced to normal. Reversing is effected by the same sequence to the left.

Engine Structure.

In single piston engines the fixed portions from the cylinder cover down to the bedplate, have to stand the main combustion load, and it should be the object of the design to take the load as directly as possible to the bedplate. If saving of weight was the only consideration, then the adoption of through bolts from the cylinder head to the bedplate, would be the one solution. However, the question of rigidity of the structure, makes the employment of some form of strong crankcase construction desirable. On examination of the engine structure in most types of single piston engines, one notices a similarity in design, which goes to bear out the statement that ultimately a very limited number of designs will survive.

A. columns over the main bearing either joined to the next as one casting, or separate, with facings provided so that the closing in doors may be oil tight, is the general practice.

Associated with the design of the crankcase is that of the engine crosshead, in so far as providing guide plates for the slippers is concerned.

Fig. 20 shows the two designs of slipper arrangements generally adopted. The divided slipper arrangement as adopted by F.I.A.T., Sulzer and Scott-Still, has advantages of access to the crankcase over the single slipper arrangement. The through bolts taking the load from the cylinder head to the bedplate may alternatively be taken to the top of the A columns, and then *via* the columns to the bedplate. The latter is the method adopted in the Scott-Still engine.

As far as possible, absolute freedom of expansion should be provided in the cylinder of an oil engine.

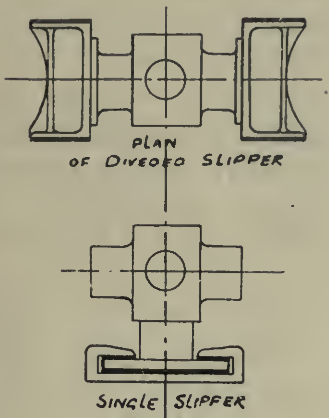


FIG. 20

Fig. 21 shows a photograph of the Scott-Still single unit engine. The steam cylinder is supported on a flange at the middle of its length, and is free to expand up and down.

The combustion cylinder is supported on the short cast iron standards seen at the top of the column, and is quite separate from the steam cylinder, except by concentric location. The combustion cylinder is free to expand through a gland in the cylinder jacket at the top, and has absolutely no axial or circumferential restraint.

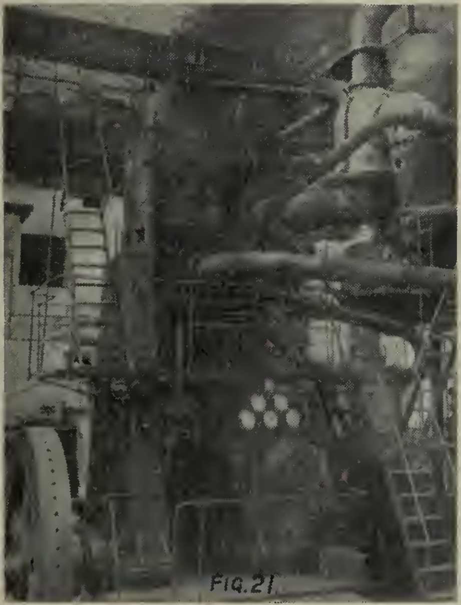


FIG. 21

The steam load is taken direct to the column from the cylinder, and the combustion load *via* the through bolts from the cast steel jacket to the columns.

In the opposed piston type of engine, the framing does not take any substantial part of the combustion load. At the same time, the question of rigidity must be considered, especially as certain thrust surfaces must be provided at the top piston, at a distance from the crankshaft which might give rise to awkward movement.

The dynamic balance of the engine is reputed to be perfect.

The Cammell Laird Fullagar engine represents a very neat form of opposed piston engines. By attaching the upper piston of one line to the lower piston of the adjacent line, a reduction of the cranks of from 6 to 2 is obtained in a two line engine. Additional guide surface is required to look after the thrust of the cross tie rods.

This combination of cylinders and pistons has made the opposed piston type of engine a very promising one for large powers. The low piston speed and revolutions obtained in these engines should appeal to every marine engineer..

Conclusion.

As is obvious from the wide nature of the subject chosen for this paper, only very brief references to certain features of design were permissible, and much has been left out on which a discussion would be extremely valuable.

Such matters as lubrication and ship arrangements have not been referred to.

No reference has been made to the double acting Diesel engine, although it is well known that several firms have experimental units in hand.

The Chairman: I would like to say that the outstanding feature of this paper is its fairness. It deals with a very interesting phase of this subject, the Scott-Still engine. I have only one question to raise—in one of the photographs a part of the apparatus was referred to by Mr. Hutchison as a regenerator. Actually it is a Cochran boiler, but it is not clear to me whether it is a boiler as such in which fuel is burnt or whether the exhaust gases only are used in it. I would also refer to Sir Dugald Clerk's remarks concerning this engine, in which he stated that the logical outcome of the Still idea would be the use of the exhaust heat, and the heat of the jacket, in a turbine engine. I would like the author to say a few words on Sir Dugald Clerk's proposal to use the jacket water in a boiler to make steam for a low pressure turbine.

A Member: I would like to ask what is the weight of the Scott-Still engine as compared with a Diesel engine of the same power, also a reciprocating steam engine, and a steam turbine, geared or ungeared. No doubt the principle of the Still engine is right, but the photographs have shown that there are many complications, which would be a serious drawback in the case of a set of marine engines.

A Member: I would like to supplement the remarks of the previous speaker by noting that Messrs. Alfred Holt and Co. are displaying much enterprise and have taken the lead, as far as this engine is concerned, by placing an order for an installation on the Scott-Still system in a vessel of 400 ft. length, 52 ft. beam, and 2,500 B.H.P. This is the first order for a motorship with this class of machinery. I understand that the guaranteed fuel consumption will be ·375 lbs. per B.H.P., with solid fuel injection. There is much to be said for Messrs. Holt's enterprise. They have been very progressive in the past. There is one point on which Mr. Hutchison touched, the method of injecting the scavenging air, in connection with which I am inclined to think the designers have not altogether allowed for one factor affecting marine work. Where you have air varying in density in different parts of the world, due to tropical temperatures, etc., it seems that difficulties may arise. Diagram No. 13, with the constant lead, I think it was termed, reminds me very much of Stevenson's link gear and Allen's link gear. Some of us present will remember the hard driving required with these link gears, and I notice the conditions here are somewhat similar. I notice the Scott-Still engine has a cast-iron liner; I suggest it is necessary to be very careful with such a liner if the super heat is more than 100° F., having in mind Professor Carpenter's experiments which proved that cast iron grows under such conditions. Also you may find pitting occur; I have had experience of this due to the scouring action of the gases. With reference to the Cochran boiler, I should like more information regarding the procedure under starting conditions—is the boiler fired at the outset, and the steam generated afterwards by the exhaust gases? Another point of importance to marine engineers—can the space required by this type of engine be kept within the limits of the ordinary engine room space of a ship, in view of the enlarged cylinders, crank cases, etc.?

A Visitor: May I ask whether with the Still system, owing to the higher cylinder jacket temperature, a greater M.E.P. is obtainable. This point is very interesting to me, as although I am not at the moment directly concerned with the Diesel engine, I did take a considerable interest, and was in a way, connected with some of the early work in connection with Marine Diesels.

Because of this it is somewhat disappointing to me to find that the M.E.P. of the Diesel has not increased in the same ratio as the increase that has been obtained with the petrol engine, and as a heat engine, I am afraid that the Diesel has allowed the Petrol engine to get ahead.

It would also be interesting to know the weight of this engine, my requirements I may add would be met by an engine of say two hundred H.P. weighing about fifty pounds per horse power.

I am glad to note that in the Still Engine, blast air has been dispensed with, this is a point that I laboured in the early days, but at that time one could not get makers to realise that high pressures were necessary. Three or four thousand pounds were considered sufficient, and even five hundred pounds in the case of the large Semi Diesels, and I tried in vain to get designers to try a much higher pressure. I understand that present practice goes up to ten thousand pounds, and I should be glad if the author could let me know if experience has shown that there is any limit, above which no benefit is derived from any further increase.

With regard to the Still Fuel pumping system, it contains a very excellent feature in eliminating any possible pressure on the fuel valve till that pressure is actually required. I cannot help thinking however that this system is far from ideal. I should like to see the differential valve abolished and the fuel pumped direct into the cylinder by a pump that attains the full fluid pressure at once, sustains it during the fuel injection period and allows the pressure to immediately fall to zero. Such a pump can be produced I am sure.

The Chairman: Before I call upon the author I will only observe that the makers of the Scott-Still engine claim a very low consumption. I presume that the figures quoted are laboratory figures, and we have learned from experience that laboratory results require modification under practical working conditions. The figures are allowed to be a little flattering; I cannot help noticing that they are all about ·42 or below. In the log books the figure averages ·44. I suggest there is a considerable margin between the laboratory figures and those obtained from practical running.

The Author: The function of the regenerator is to take the heat out of the exhaust gases. In the case of the experimental unit, a Cochran boiler is used, but for marine purposes it is proposed to use a Yarrow type boiler as a steam collector and final regenerator. A small tubular regenerator is placed close to each cylinder. With regard to Sir Dugald Clerk's proposal as to the use of low pressure steam, it is convenient to use the under side of the main oil piston as a steam cylinder and a reasonable steam consumption will be obtained in a multi-cylinder engine using high pressure steam of about 120 lbs. per square inch.

If at any time the Still engine is made double acting on the oil side then the steam would be used in a separate engine. Regarding the question as to whether the steam should be used at a low pressure, that would mean low temperature in the cylinder jacket resulting in lower combustion efficiency and probably higher compression pressure. Regarding the point raised that the figures obtained from the Scott-Still engine are laboratory figures; the tests of the single cylinder engine were made over long periods and under ordinary conditions. On the steam side, as can be seen from the saturation line shown on the H.P. card in Fig. 5, the diagram factor is poor.

It is anticipated that in a ship where the H.P. cylinder would be the under side of one of the main oil pistons the diagram factor would be 80%, and the fuel consumption would go down to about ·36 lbs. per B.H.P. hour in consequence. Capt. H. Riall Sankey's report on this engine published in "Engineering" of the 10th February refers to this matter. Diagram A shows the function of the regenerator or boiler. It is fixed at such a height that the water level in the boiler is higher than the cylinder jackets. The exhaust gas temperature is reduced from 700° F. to 180° F. As regards the method of raising steam; an oil burner is provided in the boiler for raising steam from cold. The burner is shut off when the engine is on load on fuel.

The question of weight has been raised. In a paper by Mr. Rennie comparative figures are given, if we take an example—reciprocating steam engines and Still engines of 1,300 H.P., the weight of steam reciprocating machinery is 338 tons, as against 315 tons for the Still engine installation. The weight in lbs. per B.H.P. is about 210.

A Member enquires about the effect of temperature of the scavenge air under tropical conditions. As in the case of aero engines at high altitudes there is reduction of oxygen supplied to the cylinder each stroke, reducing the power capacity of the engine and increasing the fuel consumption for a given power. In the Still engine less radiation would then take place from the jackets, etc., giving a slight increase in steam.

Regarding the question of growth of the cast iron, the iron used for the cylinder has been subjected to growth tests in our works laboratory. The results indicate that the iron used does not grow to any serious extent.

Regarding the space occupied by Still engine machinery, on the average it is about 10 to 20% less than for reciprocating steam engines or turbine installations.

A question raised by a visitor is how high the mean indicated pressure may be raised. This is not easy to answer directly.

In the experimental Scott-Still engine the fuel consumption is a minimum at about 75 to 80 lbs. per square inch as shown in Fig. 7. Under sea conditions it is desirable to have a wide margin of power to cope with emergencies, and it is not wise to rate marine engines too highly.

The effect of the hot jacket on the mean indicated pressure should enable a higher M.I.P. to be carried due to increased combustion efficiency. A loss may occur due to reduced charge weight. In any case any gain in maximum M.I.P. will not be very great. The maximum M.I.P. depends on the complete combustion of all the air and oil passing through the engine.

Regarding the question of fuel injection pressure; very little gain results from pressures over 6,000 lbs. per square inch, although this depends on the proportions of the spray cap and combustion chamber shape.

The Chairman: What temperature do you carry in the jackets?

The Author: About 340° F.

The Chairman: Will that affect the liners in the larger engines? What is the largest liner you have tried?

The Author: So far, 22 in. diameter. By reinforcing, a very thin wall can be used and a high jacket temperature used.

The Chairman: What pressure have you in the jackets?

The Author: 120 lbs. per square inch.

The Chairman: That seems a good idea, as it reinforces the liner.

*Parts of the Diesel Engine.

BY C. N. HUNTER.

It is not the intention of the writer to give a theoretical treatise on the Diesel Engine, but rather to describe some of the important parts, and to give an impression of the engine's behaviour at sea in all conditions, and to show that this new Prime Mover for ship propulsion is an accomplished fact with a great future. There are several types, *e.g.*, 4 cycle single acting, which is truly the "Diesel"; the 4 cycle double acting; 2 cycle single or double acting; 2 cycle with opposed pistons in the one cylinder having 3 cranks to each cylinder; and the 2 cycle opposed pistons with the top pistons connected to the adjacent cylinder's bottom piston (in this engine there is one crank for each cylinder). To give a description of each engine would take up more time than is desired, and I intend to confine my remarks generally to the 4 cycle engine with blast injection, as it is from the experience gained with this type that the others have been developed.

About 1912 a certain Continental steamship company converted a number of their vessels from steam to "Diesel," and this proving satisfactory, both from an economical and practical point, they decided to have their remaining steam vessels sold or converted. The company has now a fleet of Diesel engined vessels, which is stated to give entire satisfaction, thus shewing that the Diesel engined vessel was and is a proposition worthy of further study.

There are at present some 200 vessels building of 3000 tons D.W. capacity and over, to be equipped with Diesel engines, and of these the 4 cycle strongly predominates.

In this country several shipowners have Diesel engined vessels running with entire success.

The principle on which the engine works is generally known, but for the purpose of explanation, the following will perhaps be of assistance.

Commencing with the piston on the top, the 4 cycle is as follows:—

First Downstroke.—The inlet valve opens and air is drawn into the cylinder at atmospheric pressure, through a slotted pipe, which is open to the engine room.

Second Stroke.—The inlet valve closes and the air is compressed on the return stroke of the piston to about 500 lbs. per square inch with a temperature of 1200° F. or more.

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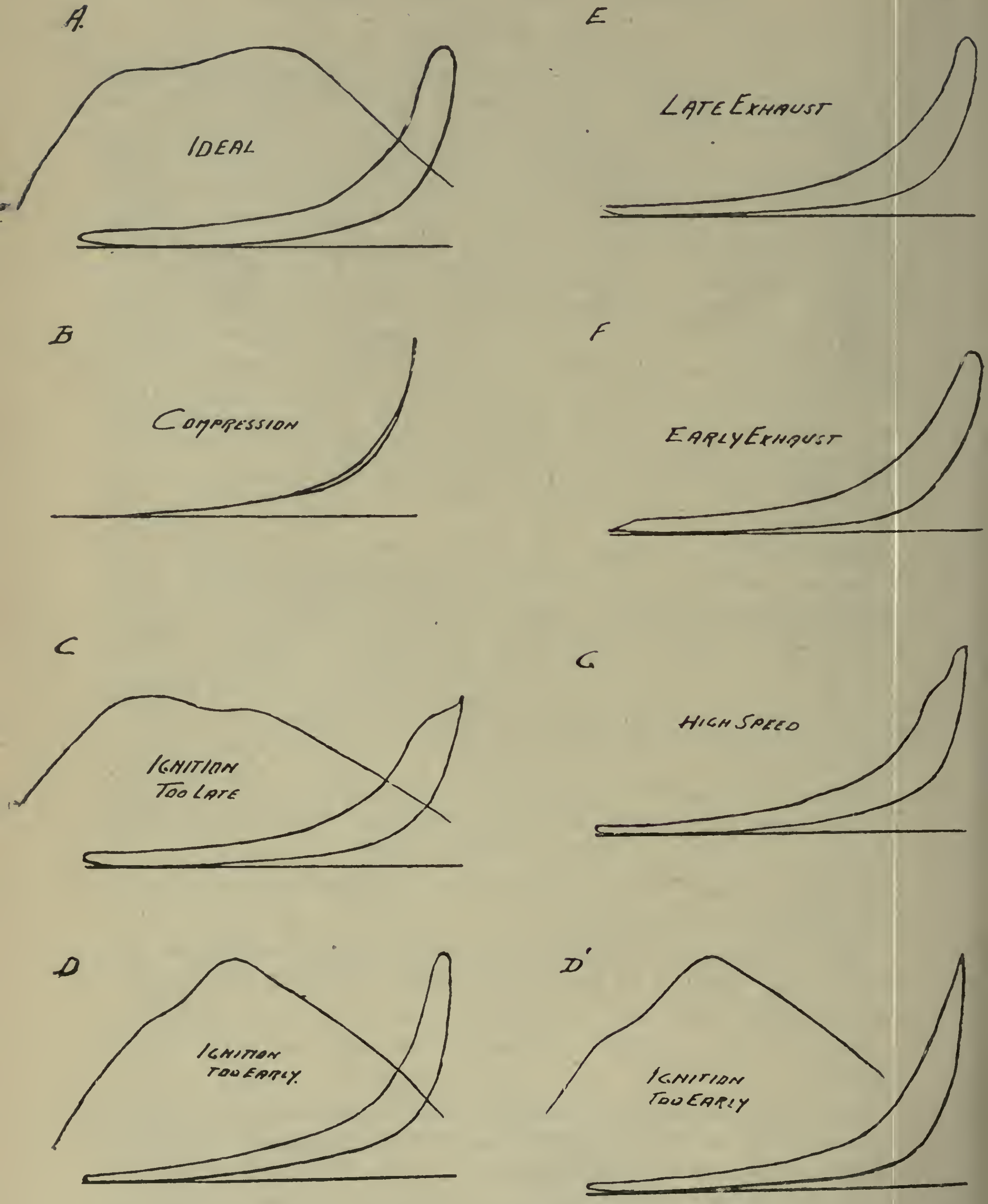

A.
IDEAL
E
LATE EXHAUST
B
COMPRESSION
F
EARLY EXHAUST
C
IGNITION TOO LATE
G
HIGH SPEED
D
IGNITION TOO EARLY.
D'
IGNITION TOO EARLY

FIG. 1

Third Stroke.—Fuel is injected into the cylinder by compressed air at pressures varying from 42 to 65 atmospheres, according to fuel used and speed required, and, mixing with the compressed air, ignites; fuel is cut off at about 1/10th of the stroke, combustion takes place and expansion follows until the end of the stroke.

Fourth Stroke.—The exhaust valve opens and the exhaust gases are expelled from the cylinder.

Fig. 1 is a typical 4 cycle indicator diagram.

Figs. 2 and 3 are respectively diagrammatic sketches of the 1913 and 1920 arrangements. It will be observed that many alterations have taken place during that period.

I will now endeavour to explain how some of these alterations came about.

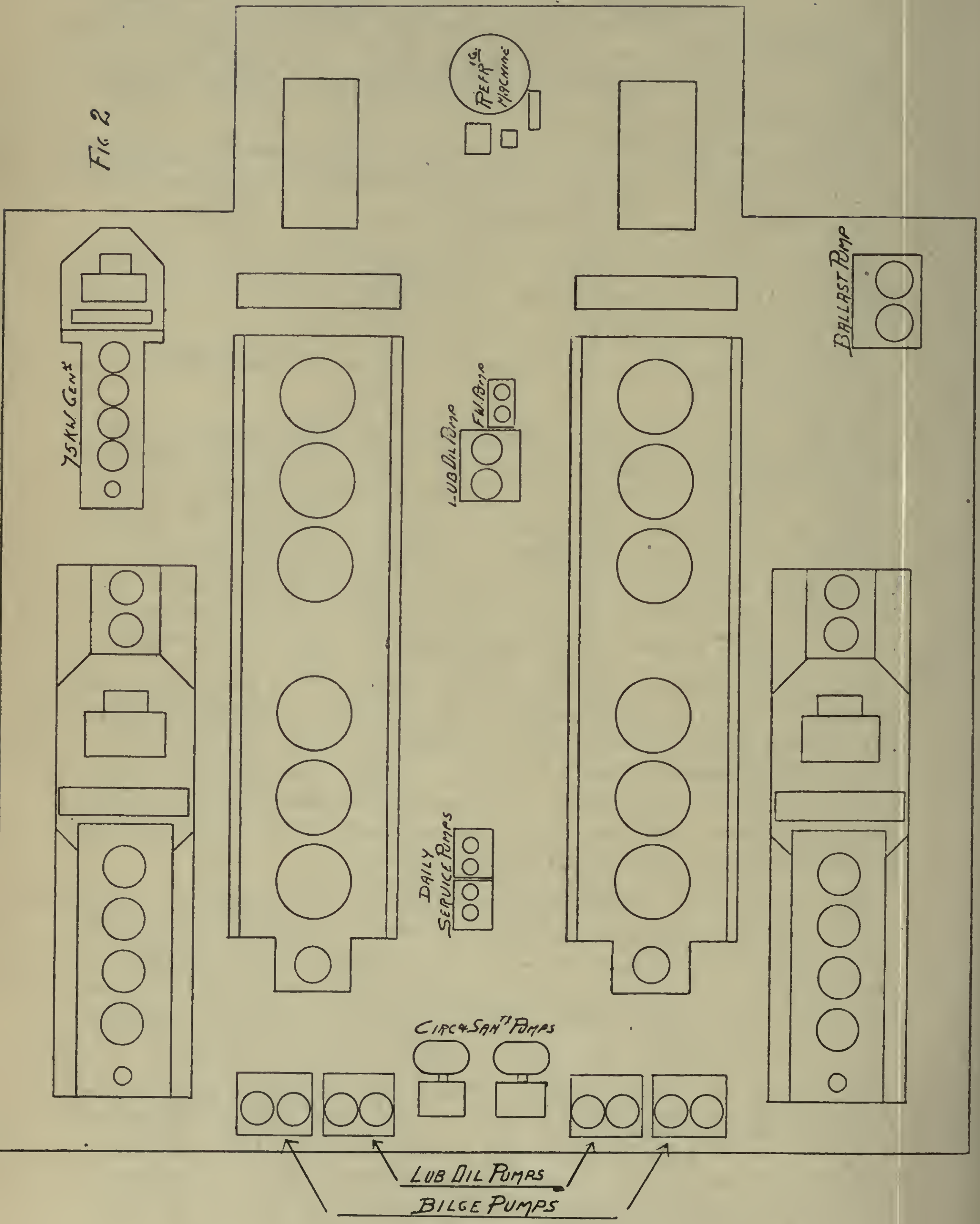
In Fig. 2, the main auxiliaries consisted of two sets of 4 cycle 80 H.P. cylinder engines coupled to a dynamo and an air compressor, the dynamo generating power for driving pumps, steering gear, etc., and the compressor supplying air for main engine purposes. As it was necessary to use one of the dynamos at all times both at sea and in port, it will be observed that there was little time for overhauling and unless the Diesel engine is maintained in good condition serious troubles are liable to develop.

This arrangement proving unsatisfactory, the next step was to build the units separately, *i.e.*, by having an engine to drive the auxiliary compressors and an engine for each dynamo. The final arrangement is shewn in Fig. 3, and has proved highly satisfactory. The design of the auxiliary remains much the same, but now that the units are separate, 3/50 engines have superseded the 4/80 for driving the generators.

The 4/80 engine piston requires artificial cooling (it is usual to oil cool the piston on this size of engine), whereas with the 4/50 piston this is not necessary, as sufficient cooling is effected through the medium of the cylinder liner, round which the cooling water is circulated, and by introducing the smaller cylinder considerable economy is effected in lubricating oil.

In Fig. 2 the main engine is fitted with a single stage compressor, and its suction is coupled to the reservoirs, which are maintained at a pressure of 20 atmospheres by the auxiliary compressors, so that the main compressor drew in air at 20 atmospheres, and increased the pressure to 60 atmospheres for blast purposes. The main pistons were oil cooled.

Fig 2


REFR'G MACHINE
75 KW GEN'T
BALLAST PUMP
LUB OIL PUMP
F.W. PUMP
DAILY SERVICE PUMPS
CIRC & SAN'Y PUMPS
LUB OIL PUMPS
BILGE PUMPS

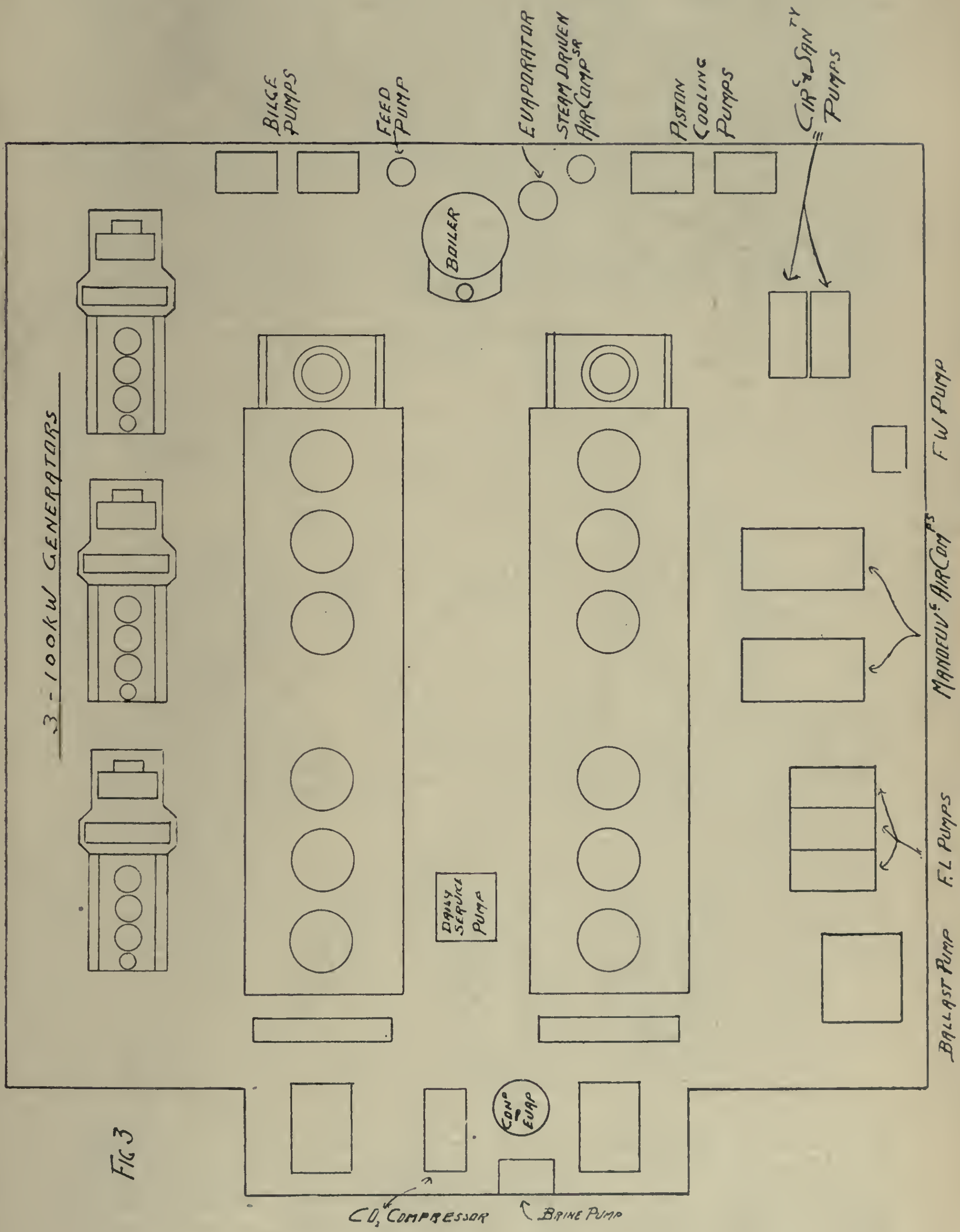

BILGE PUMPS
FEED PUMP
EVAPORATOR
STEAM DRIVEN AIR COMP^SR
PISTON COOLING PUMPS
CIR^G & SAN^TY PUMPS
BOILER
3-100KW GENERATORS
FW PUMP
MANEUV^G AIR COMP^RS
F.L PUMPS
BALLAST PUMP
DAILY SERVICE PUMP
CON^D EVAP
CO2 COMPRESSOR
BRINE PUMP

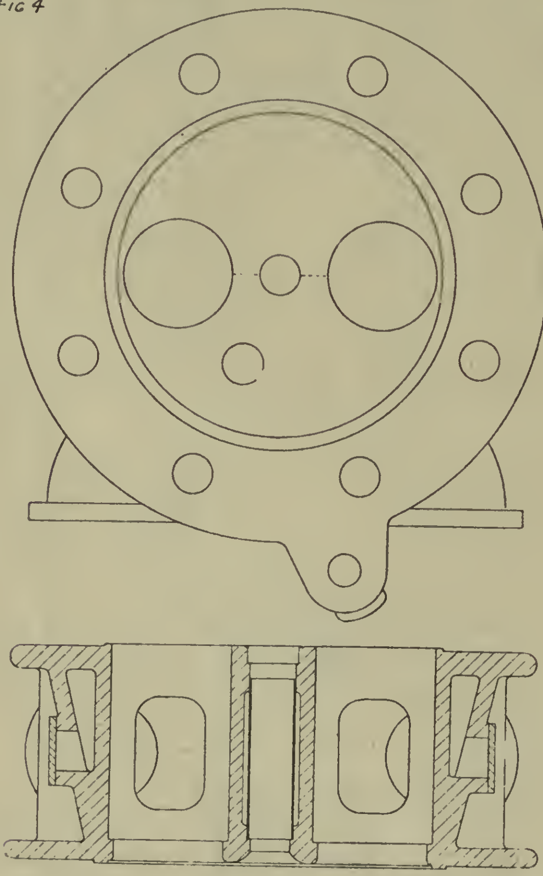
FIG 3

In Fig. 3 the main compressor is 3-stage, and is not dependent on the auxiliary. The main pistons are fresh water cooled. The auxiliary compressors are motor driven, and are used only when manœuvring. In both cases all the pumps, steering gear, and refrigerating machine are electrically driven.

Cylinder Covers.—The suitable design of the cylinder cover gives rise to considerable difficulty and anxiety, as it contains all the valves and water passages. It is a difficult casting to make, and is usually of cast iron. The cover is water cooled, but as the water has rather an obstructed passage this has resulted in a number of cracked covers. The cracks invariably extend across the bottom face, from exhaust to inlet valve space, and in some instances up the exhaust valve passage. It is claimed that these cracks occur in starting up. Attempts have been made to weld these defects, but with uncertain results. An arrangement for steam heating the cover before starting is being

CYLINDER COVER.

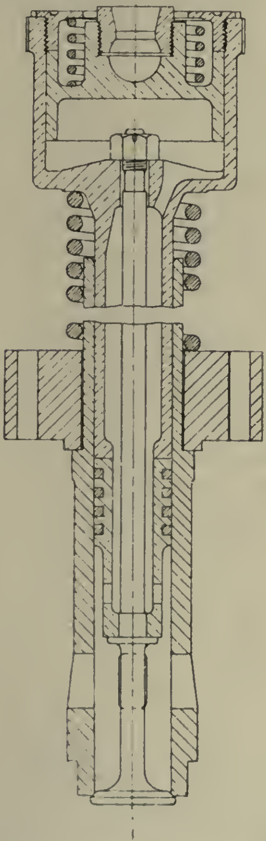
Fig 4



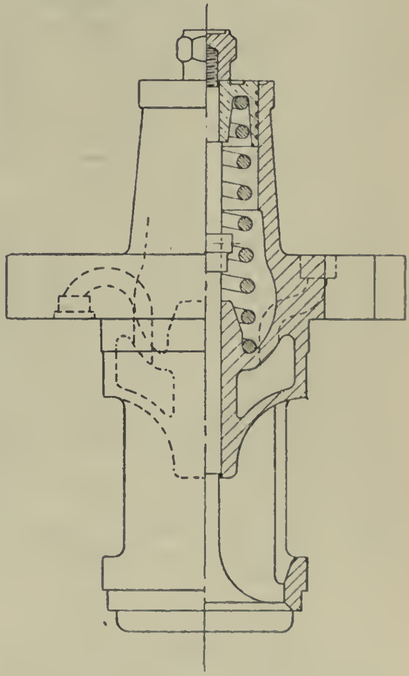
experimented with, and up to date no covers have been reported cracked where this system has been adopted. The bottom face of the cover undergoes a rapid change of temperature when starting up from cold, say for instance, the temperature of the engine-room is 60° F., air is drawn into the cylinder and compressed to about 1200° F., when the fuel is injected the temperature is further increased to approximately 2000° F., so that it is not surprising when a cracked cover is found.

There are five valves in the cover: the starting, the exhaust, the inlet, the fuel and the escape valve. The only difference in the exhaust and inlet valves is that the exhaust valve (Fig. 5) is water cooled, whereas the inlet is not. A starting valve is shown in Fig. 6, and a fuel valve in Fig. 7.

STARTING VALVE.
FIG 6



EXHAUST VALVE.
FIG 5.

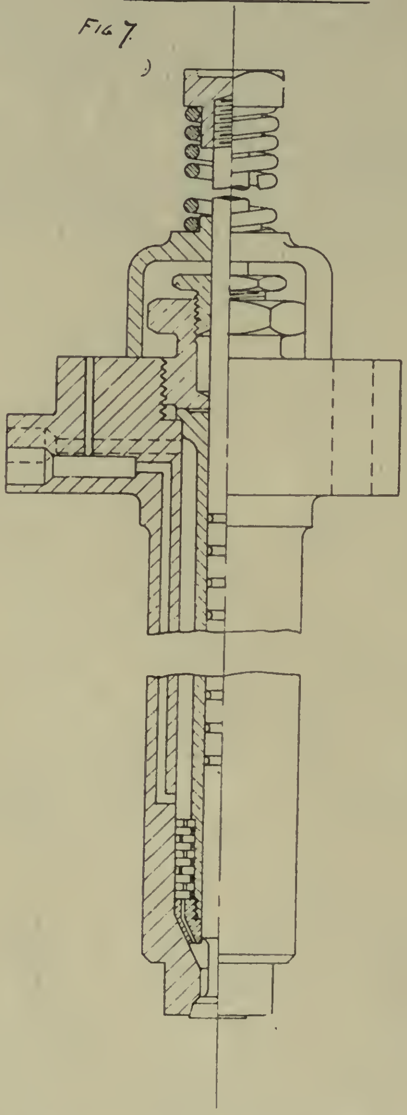


Starting Valves were the cause of a considerable amount of trouble before rings were substituted for soft packing. These valves, if overhauled about once in 12 months, should give no cause for complaint.

Great care must be exercised when jointing the valve to the cover. The two joint faces must be perfectly tight for this reason: if the bottom joint is not tight, in the firing and exhaust stroke the live and exhaust gases will pass the joint, and thereby heat up the starting air line and fill the piping with a carbonaceous deposit, which will clog up the leaky valve and finally pass through the starting slide, and leave this important valve in very bad working condition, so that when a manœuvre is attempted trouble is likely to ensue. A serious accident can

FUEL VALVE.

FIG 7.



result from a leaky top joint, and in fact did happen recently on a vessel, which, after running about 4,000 miles, was stopped to receive a pilot. On restarting an explosion occurred in the starting air line which wrecked it and also a compressor head to which it was coupled.

On the voyage the fuel valve gland had leaked and the fuel ran on to the cover and under the starting valve joint into the starting air line; when the engine was started one of the starting valves stuck open, with the result that this cylinder fired past the starting valve, and the flame ignited the fuel lying in the air line, thereby causing the disaster. Therefore, too great care cannot be taken in jointing these valves.

Exhaust Valves require attention more frequently. The valve face and seat are detachable; the main engine valves are made of hard close-grained cast iron, suitable for high temperatures.

An accident with an exhaust valve, which came under the writer's notice, may be of interest. The engine driving a generator had been at rest for a few hours after a long run, and on restarting a sudden heavy jolt in one of the cylinders was the only result. The exhaust valve was taken out and the cylinder found to be full of water. On further examination the connecting rod was found to be bent, and the bottom part of the liner broken off. The cause of this catastrophe lay with the exhaust valve. Corrosion had been going on inside the water space, and finally the water found its way through, and as this valve remains open when the engine is in starting position, the water gathered on the top of the piston, filling the cylinder, so that when an attempt was made to start the engine the result was as already stated. As there are no means of inspecting this water space, I am of opinion that if this space were tested by pressure occasionally, accidents such as mentioned would be prevented.

Fuel Injection Valve.—The assembling of this valve requires special attention when adjusting the pulveriser, packing, and fitting of the non-return valve.

The pulveriser consists of a number of thin steel discs with slots or holes drilled around the outer edge. The number of these discs depends on the viscosity of the fuel. The non-return valve is fitted immediately above the fuel entrance to the pulveriser, and is placed there to prevent the blast air passing back into the fuel pipes. There is also another non-return valve placed in the fuel pipe between the fuel valve and the fuel pump.

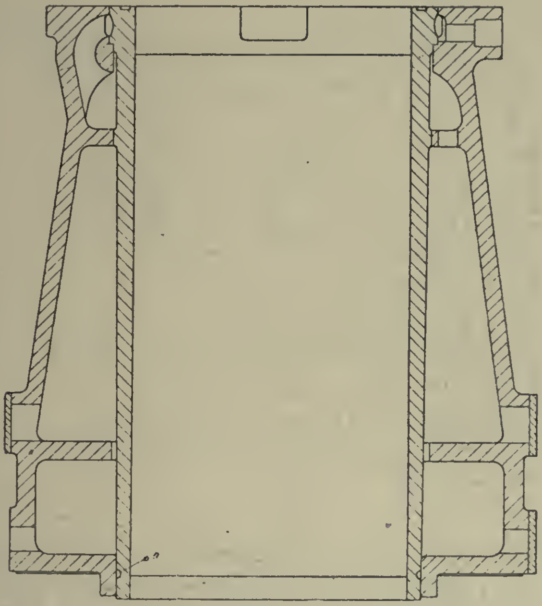
This precaution is taken to prevent the fuel pump becoming air locked in the event of the top valve leaking. A fuel valve may run for several weeks or may not last an hour. If the fuel is not properly filtered and a particle of grit adheres to the valve face, the flame will burn the valve head and seat, with the result that there will be a loss of blast air, which will necessitate the stopping of the engine in order to replace the valve. The valve spindle frequently becomes pitted in way of the packing, due to impurities, grit or water in the fuel. The spindles when in this condition seize on the packing and remain open during the firing stroke, resulting in the same trouble as in the previous case. The valves are liable to be burnt if the engine is kept running too slow for any length of time. It must be remembered that when running slow the valve has the same lift as when running at full speed, and therefore, as there is less fuel being delivered, more blast air is used. The valve is also moving slower, and exposed to the flame for a longer period. These are the general fuel valve defects. Some engines have been designed with a variable fuel valve lift according to speed. This, in my opinion, is a very good arrangement, although it adds to the complication of the valve gear. The advantage of it is that blast air is saved, and the engine can be run much slower than in the usual way, without fear of burning the valve.

The Escape Valves are of the ordinary spring-loaded type, and must be examined periodically, as the apertures to the cover leading to the valve frequently become choked with carbonised oil, thereby causing the valve to become sluggish or inoperative. An escape valve in this condition would lead to serious consequences in the event of the cylinders becoming overcharged beyond the safe limit.

CYLINDER LINERS.—Cylinder liners are made of special hard close-grained cast iron of high tensile strength, which will give a hard and smooth wearing surface. Fig. 8 shows a liner and its attachment to the cylinder. At the top the liner flange rests in the cylinder. Great care in the design and also the fitting of this joint must be observed in view of the heavy stresses exercised upon it when jointing the cover. The bottom of the liner is a working fit into the cylinder to allow for free expansion. The water-tight joint between the liner and the cylinder at the bottom is usually made with a rubber ring (*see* Fig. 8).

The spigot joint at A is a ground face; usually a little graphite is spread over the face before jointing. The joint if not properly made will leak, and the hot gases burn the faces, and this would necessitate the fitting of copper rings in order to make the joint tight. When this is done the piston clearance must be adjusted, and suitable liners fitted at the butt of the connecting

FIG 8 CYLINDER.



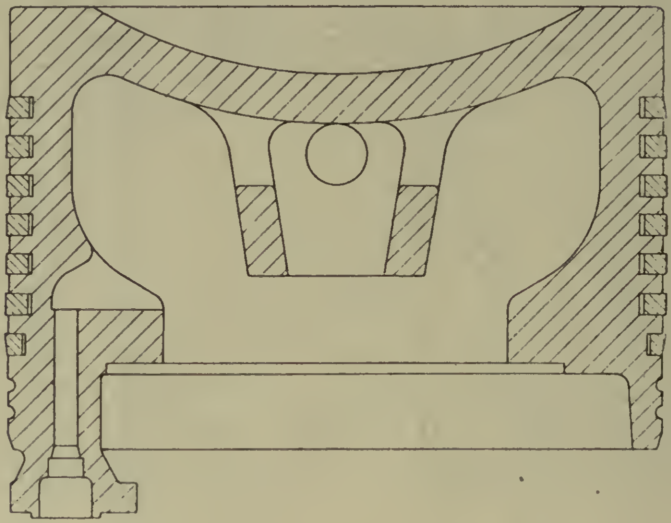
rod. The wear of the liner on an engine of the crosshead type is considerably less than that of a steam engine. Cylinder liners of engines without crossheads, but fitted with deep pistons, as is the usual practice, wear slightly oval, owing to the liner having to act as a guide. The wear on the liners in one engine of this type, after three years almost constant running, had no appreciable effect on the engine's efficiency.

A piston suitable for a cylinder 27 inches in diameter is shown in Fig. 9, and is of the water cooled variety. Air and also oil cooled pistons have been in use. The air cooled could never be considered a success. Oil cooled are still used for the smaller types, but fresh water cooling is now universally used for large marine engines.

It will be observed that the piston is dished on the crown, and is parallel at the lower point, and tapered towards the top. All Diesel engine pistons are designed with a conical top, and I think the majority of designers favour the dished top, as it strengthens the piston and allows for a certain amount of expansion which takes place.

PISTON.

FIG 9



Main engine pistons give little trouble where M.E.P. is below $6\frac{1}{2}$ Kg per sq. cm., but in the smaller engines, where this pressure is exceeded, many cracked pistons have had to be dealt with. Welding has been tried, but so far is a failure. Several pistons are now running repaired as shown in Fig. 10, giving entire satisfaction. The pistons are fitted with ramsbottom rings, usually 6 to 8 rings.

The usual practice is to double turn the rings in order to ensure a perfect fit. Air compressor piston rings are treated in a similar manner.

With the water cooled piston the most successful arrangement is with telescopic piping (*see* Fig. 12). The centre tube is made of iron and requires renewing occasionally, as it corrodes, probably through galvanic action which takes place between the brass tubes and the piston.

The discharge temperature of the circulating water should be maintained as near as possible to 160° F., as if kept below that there is a loss in efficiency, and if kept much higher the piston is liable to become steam-locked, with the possibility of a cracked piston. Due consideration must also be given to the amount of impurities in the form of lime and other matter contained in the water which are liable to deposit in the cooling water space.

METHOD OF REPAIRING A CRACKED PISTON

FIG 10

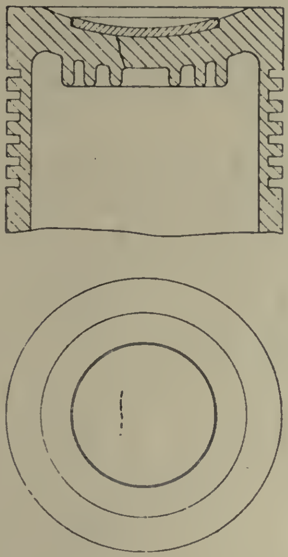
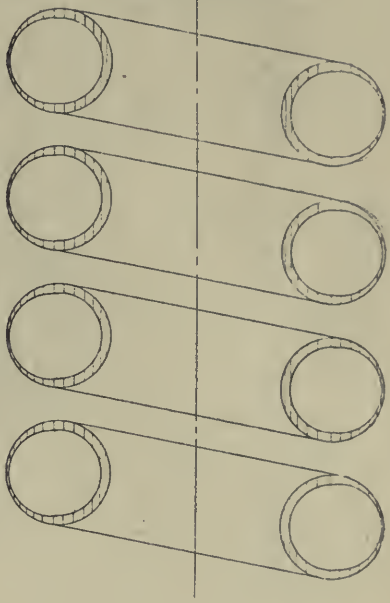


FIG 11 SECTION OF COIL.

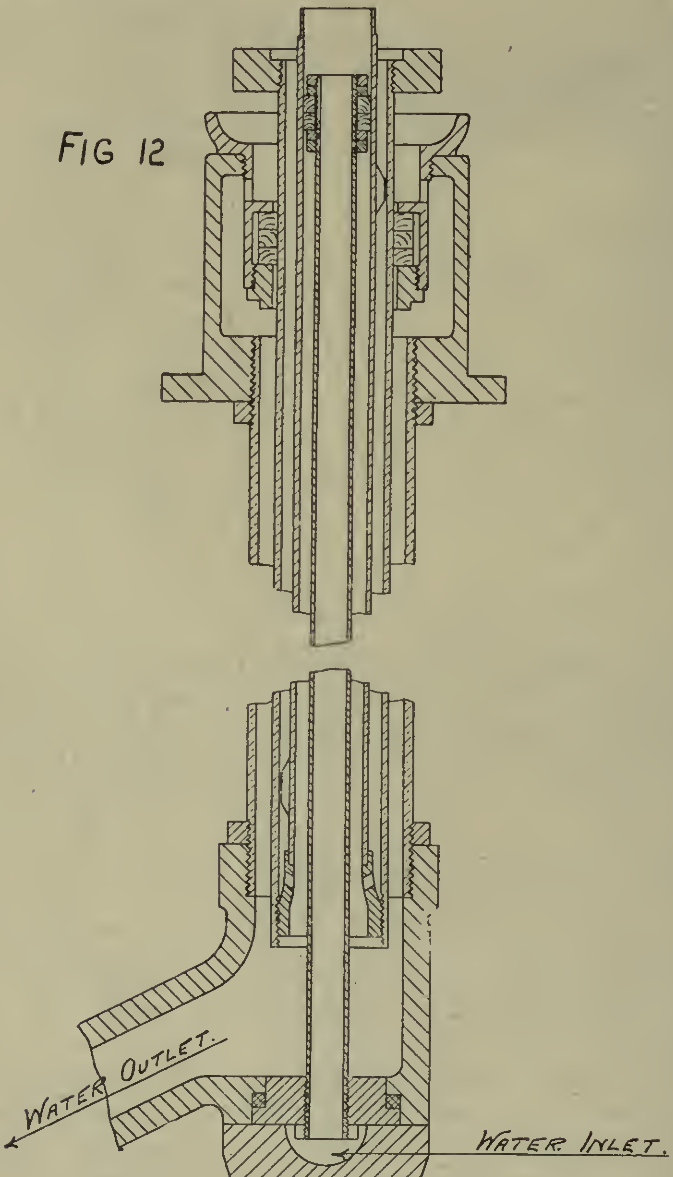


Compressors. — The pioneer marine Diesel engineers experienced no little trouble and anxiety with compressors. Many a complete stoppage was caused by a spring or valve breaking or a coil bursting. Considerable improvements have been made in recent years in air compressor design suitable for Diesel work. Main engines are now fitted with a three-stage and the smaller types with two-stage compressors.

Too liberal lubrication is the cause of many of the present day air compressor troubles. The oil becomes carbonised and adheres to the valves, causing sluggish working and inefficiency of the compressor.

When the L.P. piston is open at the bottom to the crank case, care should be taken that the rings are a good fit and the slots in the inlet pipe kept clean, otherwise the vapour from the lubricating oil in the crank case will find an entrance into the cylinder and cause trouble. On one occasion, when having trouble with a compressor, the writer noticed the receiver between the L.P. and H.P. stages blue with heat. Needless to say the engine was immediately stopped, and on opening up the compressor he found the majority of the valves practically solid with carbonised oil. The trouble was caused by the inlet pipe being partially choked and the L.P. piston passing oil vapour from the crank case.

FIG 12



PISTON COOLING WATER PIPES.

The compressor coils should be kept clean, but never heated over a fire for cleaning purposes. This also applies to the blast pipes. If heated the oil inside carbonises and forms a porous skin inside coil.

When the coil is put into use again, the action of the air passing through this porous matter overheats the coil locally and weakens it, and in a short time the coil will in all probability burst. The only method of indicating the wear on the inside of the coil is by weighing periodically, and if the weight is less than the original by a predetermined percentage the coil should be discarded.

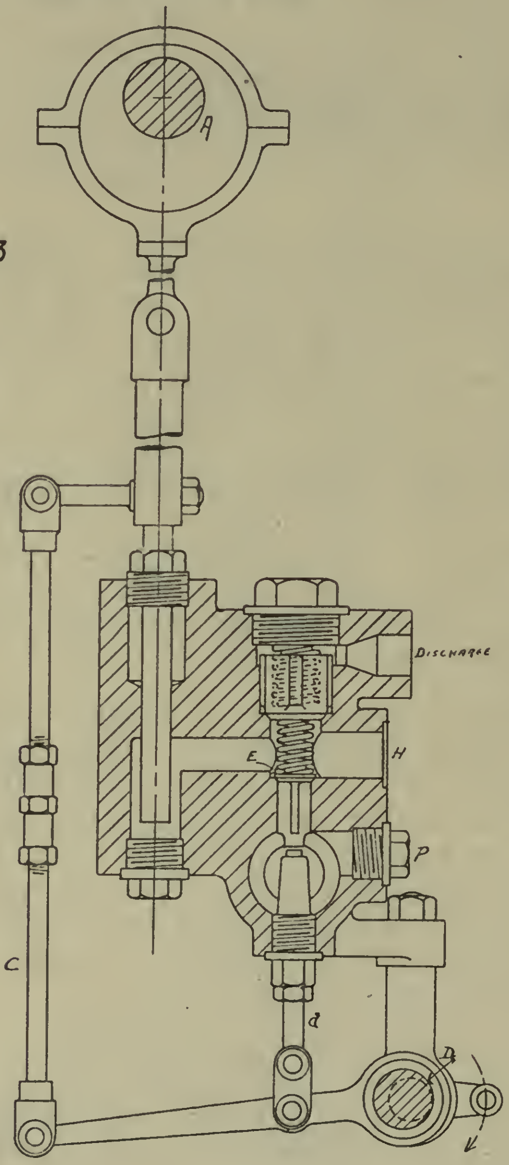
The writer had occasion to examine the remains of a compressor in which the coil had burst, and found the coil worn quite thin on the outer half of the bore, whereas the inside half retained its original thickness, *see* Fig. 11. This coil had been in use for about two and a half years, and had gradually been thinned down by the action of the air rushing round the outside half of the coil, carrying with it particles of dust and carbonised oil. The water jacket was blown to pieces, although fitted with a safety valve. The safety valves are not to be relied upon in event of an accident as stated above. The writer would suggest that if a thin disc were fitted several inches in diameter and made to blow out at a few lbs. over the maximum circulating water pressure, accidents such as described would be avoided.

Air Injection Bottles.—The air bottles are now generally made of solid drawn steel, reduced at the ends and fitted with screwed flanges. In the early days the bottle was drained through an internal pipe fitted with a valve at the top. This practice has now been discontinued, as it was found in many instances that the pipe had broken off at the neck, which made drainage impossible. These bottles should always be kept at full pressure and drained for water each watch, also cleaned out periodically and valves ground in.

The pipe arrangement for the air bottles should be so designed that in the event of loss of air from any bottle it could be recharged from any other bottle in the engine room. Loss of air is quite a common event when testing an engine after an overhaul, and the benefit of the above arrangement will readily be observed. The bottles are usually placed on the bottom platform with the lower end resting on a stool in the bilge: the bottom, being out of sight, is often out of mind.

Fuel Pump.—In some designs one pump delivers the fuel into a distribution chest from which valves and pipes are led to each fuel valve. The most favoured method is to have a pump for each cylinder, Fig. 13, as a more equal distribution of the oil is obtained; also it is then possible to cut out any one cylinder without affecting the others.

Fig 13



Fuel Pump

In the latter instance the pump delivers the quantity of fuel required for each combustion stroke to the fuel valve. The pump has also to overcome the blast air pressure always in the fuel valve, usually from 42 to 65 atmospheres. The quantity of fuel required varies with the load or speed of the engine, so that it will be readily understood the design and adjustment of the fuel pump requires no little skill. The body of the pump is of cast iron, the valves and plungers of steel, and the plunger is case-hardened. The gland is packed with suitable packing. In the past these glands have given considerable trouble, as they are so small and in most engines not very accessible. Some makers design the pump without a stuffing box. The plunger is then made a push fit in the pump chamber, and I understand that when properly fitted very good results have been obtained.

In governing the fuel delivered per stroke a very ingenious method was arrived at after considerable experimental work. The stroke of the pump remains constant, but only the quantity required, according to load or speed, passes the delivery valve; the surplus returns through the suction valve during the earlier part of the delivery stroke. The action of the pump is as follows:—

The plunger receives its motion from the eccentric sheave *A* on the cam shaft. The fuel supply is controlled by the tappit spindle *d* operated by an adjustable rod *c*, the suction valve *e* being held off its seat during the required portion of the delivery stroke. The control lever of the engine is connected to a small spindle on which is mounted the eccentric sheaves *D*.

If the control lever be moved in the direction of the arrow, the tappit spindle *d* is lowered, and if moved in the opposite direction the tappit is raised. *H* is a hand pump for priming the pump, and *P* is the inspection plug.

When adjusting the pump the plunger is put on bottom centre, starting lever slightly back from starting position; a clearance of about 4/1000 is then made between the bottom of the suction valve and the tappit *d* by the adjusting nuts at *C*. The mechanism of the pump is so arranged that when the engine is running the tappit spindle *d* permits the suction valve to close in time to allow the required amount of fuel to be injected into the fuel valve.

Lubrication.—The majority of Diesel engine builders have adopted an oil circulation system for the lubrication of shafting, etc., with enclosed crank chambers. The oil is delivered by a pump to the bearings at a pressure ranging from 8 to 12 lbs., and after passing through the bearings it drains from the crank pits into a tank from which it is again taken by the oil pump and circulated through the bearings. A system of filtering and cooling is fitted in the circuit, so that the oil is cleaned and cooled continuously while the engine is running. Various methods are employed to convey the oil to each separate bearing connected to the system. Some makers take a separate lead to each bearing, and others feed the oil through the hollow crank shaft. Some makers consider it unnecessary to cut oilways in the bearings, and it is claimed that the oil under pressure finds its way round the bearing quite effectively. This idea sounds good and appears efficient, but unless the liners in the bearings are a perfect fit and come up almost against the journal, and the bearings kept closely adjusted, this method is rather uncertain. A few slack bearings in the engine have quite an appreciable effect on the lubricating oil pressure.

The circulation system involves the use of a pure mineral lubricating oil having a good quality of separation from water or other impurities collected on its passage round the system. A compounded oil or a mineral oil with a natural tendency towards emulsification must not be used, as the thickening due to emulsification would in time impair the free flow of the lubricant through the piping. This feature is most important and an emulsification test should always be made before a new or unknown oil is used. Bearing this in mind an oil of the following tests will give good results:—

Sp. Gr. ·900. Open flash test 400° F. Redwood viscosity at 70° F. about 600 seconds; at 125° F. about 135 seconds.

Care should be taken in designing a circulating system to make provision for keeping the oil in good condition. In addition to the filtering mentioned, the oil pump suction should be placed at a sufficient height above the bottom of the drain tank to allow a space for separated water or dirt to accumulate without being continuously drawn back into circulation. Provision should be made to allow for the removal of water or sludge from the bottom of the tank without emptying the system.

The lubrication of Diesel engine cylinders is usually effected by force pumps, preferably sight feed, operated by the cam shaft or other moving parts of the engine. The oil is led to the cylinder walls through openings from a belt cast round the cylinder or by separate pipe connections passing through the water jacket. These openings are placed at a point between the piston rings when the piston is at the bottom of its stroke. It is desirable to have separate control over each inlet to the cylinder walls, and preferably each cylinder with its own pump, so that the supply to any cylinder can be altered independently of the others. The scraper rings are arranged on the pistons so that the oil is carried backwards and forwards with it. An oil of the physical tests previously given for bearing lubrication will also give satisfaction in the cylinders of small engines, but with large pistons it is sometimes necessary to use an oil of greater viscosity. An oil of similar Sp. Gr. and F.P. to above but with Redwood's viscosity at 70° F. about 1,300 seconds; and at 120° F. about 260 seconds will give the heavier body required.

Governing.—The governing of the main engine is quite a simple matter. The governor most suitable is of the inertia type; usually an "Aspinall" is fitted. Its action is such that should the engine race, the suction valves of the fuel pump are held off their seats by the action of the levers attached to the governor when in action. The problem of governing the auxiliary engines driving dynamos or compressors is a little more complicated. The device commonly used on engines having multiple cylinders is such that it enables one or more cylinders to be successively cut out when the load falls to a certain predetermined point. The gear so acts that these cylinders cannot resume their share of the load unless the load should again rise above another predetermined higher point. The above mentioned mechanism is attached to the governor which is of the centrifugal type, driven by means of bevelled gearing from the crankshaft.

General.—To obtain the best results from the engine, the valves must be timed correctly, and the piston clearance adjusted exactly. A slight error in any of these adjustments will make quite an appreciable difference in the engine running. For instance, if the piston clearance is not correct, the compression will be either too high or low. If too high the excessive temperatures and pressures created will cause the engine to knock heavily. If too low, incomplete combustion, a smoky exhaust, and probably a burnt exhaust valve will follow. Then again, the fuel valve if opening early will cause pre-ignition and heavy knocking; if late, combustion will not be complete; there will be a smoky exhaust, and probably a burnt exhaust valve. In turning the engine, the colour and sound of the exhaust and the general tone and feel of the engine should be carefully observed.

From the indicator diagram, as in the steam engine, sufficient information is obtainable to enable the engineer to adjust the engine to run effectively and economically. Two types of diagrams are taken, the mechanically operated and the hand card. The hand card is used for finding the compression and approximately the point of admission of the fuel.

Fig. *a* represents a typical compression card, taken when the fuel and blast air are momentarily shut off. The compression is adjusted to between 460 and 500 lbs.

Fig. *b* represents late admission with a corresponding hand card.

Fig. *c* represents early admission with a corresponding hand card.

Fig. *d* represents normal exhaust.

Fig. *e* represents early exhaust.

Fig. *f* represents late exhaust.

Slight errors shown in the cards can usually be rectified when the engine is running. It does not necessarily follow that the ideal card shown is the best card for all Diesel engines. The shape of the card varies with the type and speed of the engine, so that the card really should only be used to assist natural observation.

Fig. *g* is a diagram taken from an engine running at 450 revs. In trying to adjust the shape of this card to coincide with the ideal diagram the revolutions could not be maintained.

The alignment of shafting is much more important with the Diesel than the steam engine. Experience teaches us that the wear down is not the same on all the bearings; therefore greater care is necessary when adjusting to ensure true alignment. The usual method of taking the wear down is by the Bridge gauge, with which you are all familiar. A method adopted by engineers on several land installations is to micrometer the thickness of the bottom half of the bearings after bedding the shaft in the usual way; then at subsequent examinations the bottom halves are again measured; an absolutely correct reading is thus obtained, and adjustments made accordingly.

When adjusting bearings in a forced lubrication system where an annular space is cut in the bush to allow the oil to get into the hole in the journal, in order to pass through the shaft to the other bearings, it is found that some journals wear to such an extent that a ridge is formed, which fits into the annular space in the bearing and so prevents the oil getting to the journal. These ridges must be immediately dealt with, otherwise the life of the bearings will be short. The flexibility of the engine for manœuvring purposes is all that can be desired, but with all its good points careful handling is necessary. There are many apparently insignificant parts to be attended to while running, and which, if not examined frequently, may lead to serious consequences.

All valve clearances must be especially attended to, also the fuel pump, adjusting rods, levers, etc., should be examined at regular intervals. The settling tanks (there are usually two each of sufficient capacity to last 12 hours) should be drained for water and sludge, and the float tested occasionally. Pressures and temperatures should be recorded regularly, and there are quite a few, *e.g.*, blast air, starting air, jacket circulating water, piston cooling water and lubricating oil. Exhaust temperatures should be noted when possible. On some ships pyrometers are fitted in the exhaust pipe of each cylinder. The blast air pipes at the fuel valves should be felt now and then, and if any of these become hot it is a sure indication of a leaky fuel valve, a barking or grinding sound from a cylinder, invariably due to a dry piston, must be immediately attended to, as once this starts the piston soon heats up. The writer recently had an exciting few minutes with a hot piston. The automatic lubricator to the cylinder had jammed and the piston started grunting. In a few minutes it had attained a dull red heat on the one side and several red streaks showed on the liner. Some time elapsed before the piston and liner were brought back to normal again. This incident proves that very little warning is given and how quickly troubles arise in engines of this type, also that the engineer in charge must always be on the alert.

Possibilities.—The question of the ability to fit higher power engines of the internal combustion type in ships is dependent upon economic factors in the first place.

At the moment the highest power developed by such engines per cylinder is about 400 h.p. To equip a ship of 12,000 h.p. would therefore require no fewer than 30 cylinders. The prime cost and upkeep of such an outfit owing to the "multiplicity of bits" would evidently be serious factors. If, however, six cylinders only could be used, then the question comes within the realm of possibility; such an engine would require to develop 2,000 h.p. per cylinder. So far it has been quite impossible to build engines of this power, and for a very simple reason. In any internal combustion engine a definite percentage of the total heat value of the fuel consumed has to be transmitted from the burning fuel within the cylinder through the walls thereof to the cooling medium, whatever it may be, without the said cylinder.

Now the area of the internal walls of the cylinder, including its head and piston top, bear a definite relation to a liner dimension which is the diameter of the cylinder. This relation increases with increase of size as the square of the cylinder diameter, if engines of uniform proportion are being compared. The amount of fuel burned increases as the volume contained within the cylinder, or as the cube of the cylinder diameter. The percentage of heat transmitted being constant, evidently the heat transmitted per unit area of surface increases with inincrease of cylinder diameter. Some particular cylinder size will therefore be at length reached, at which the required rate of heat transmission through the cylinder walls is higher than the metal of these walls is capable of, and serious trouble will occur. Such is found to be the case.

The result in practice has been that the cylinders of large internal combustion engines have had to burn less and less fuel per unit volume as the size increases, the power only being able to increase as the square of the diameter, instead of as the cube as is theoretically possible.

A well-known Continental firm of Diesel engine makers some time ago built a cylinder to develop 2,000 h.p. The actual results were not published, but from information gathered the writer understands that this firm has now undertaken to build cylinders up to 900 h.p. each. This is a big increase from 400 h.p., and it will be interesting to watch the development of these internal combustion engines.

During a trial recently of a Doxford engine, which is of the opposed piston type, each cylinder developed 750 I.H.P. This is the highest power obtained up to the present in any completed multi-cylinder engine.

In conclusion, I should like to state that interesting developments have taken place with regard to the "Still System." To describe this engine is not within the scope of this article, but briefly, the engine is a combination of Diesel and steam. One side of the piston works on the Diesel system and the other on steam, the waste heat of the Diesel cycle being used for the generation of the steam so used. The "Still System" of cylinder for large engines can be designed with a definite rate of heat transmission per unit area, whatever the engine power; the result being that cylinders can be arranged for 2,000 or 5,000 h.p. each without any possibility of stressing the materials used in the construction higher than in quite reasonable everyday engines. The results will be lower first and upkeep costs, and reduced weight per horse-power.

SESSION 1922-23.

President : ENGINEER VICE-ADMIRAL SIR GEORGE G. GOODWIN, K.C.B., LL.D.

Marine Diesel Engines.

Running Conditions and Operating Symptoms.

BY A. J. BROWN.

Read Tuesday, December 5, 1922.

FOR the successful operation of the machinery in a motor ship, or for that matter, any ship, it is essential that the Chief Engineer should know the exact condition of each constituent part and what service it is capable of before needing overhaul. This information is invaluable, not only from the point of view of reducing the amount of work done in the E.R. and so lowering maintenance costs, but also because in the event of trouble occurring much time is saved by eliminating unlikely causes while the location of the source can be more readily effected. To achieve this knowledge may entail a certain amount of extra work at the start, but it is labour that will be amply repaid.

Considering first the main engines, some time after "Full Away" has rung at the beginning of the voyage and the engines have settled down for a run of several days, a set of indicator cards both normal and light spring should be taken, while all data relating to them should be accurately noted. After a power card has been taken from each cylinder, the blast air and fuel should be shut off from each one at a time, when a compression card can be obtained. All these cards should be carefully compared and the characteristics of each noted, when a good impression can be formed of the condition of the engine at the time.

Sometime within 24 hours of reaching port another set of cards of each type should be taken and compared carefully with the cards taken at the beginning of the trip. Weather conditions must, of course, be taken into account, and due allowance made for variation.

Again, when at sea, in the early part of the voyage, a greater number of readings should be taken of all consumptions and tank soundings, while the exact conditions prevailing with regard to weather, trim of ship, etc., should be carefully noted. By carrying out this procedure, the engineers will learn to discriminate at future times and more accurate readings will result, while less likelihood will exist of false impressions being created with regard to loss of oil and water carried in the tanks.

On arrival in port at the end of a long run, a point should be made of checking all cam roller clearances and noting what differences exist from the original settings. When comparing these settings, the two lots of indicator cards should also be referred to, when the "cause and effect" may become apparent.

At this time it is also advisable to test all the valves in the cylinder head, and the piston rings as well, to ascertain the true condition of each, and the findings should be carefully logged for future reference and comparison. This testing can be done by means of blast and starting air. The indicator cocks on all cylinders should be opened, then if blast air is admitted to the fuel injection valves while the operating gear is in the neutral position, by listening at the indicator cocks any fuel valve that is leaking can be detected by the noise of the air blowing into the cylinder. Starting air valves can be checked in a similar manner, also by admitting air to them when in the "off" position. To test the exhaust or any other valves that may be in the cylinder head, the manœuvring gear should be put in the neutral position and the indicator cock shut, then if air be admitted to the cylinder with the crank on T.D.C. by propping open the fuel injection or starting air valves, any leakage that exists can be detected by listening at (1) the air intake pipes in the case of inlet valves (2) the exhaust pipe test cocks in the case of exhaust valves (3) the escape port in the case of relief valves, while piston-ring tightness can be ascertained by listening at the bottom of each cylinder.

A certain amount of this work of testing is in the way of being additional labour, since the approximately true state of each valve can be ascertained from the indicator cards and operating symptoms. However, as a method of gaining a thorough knowledge of reading indicator cards and operating symptoms it is fully justified, while from the point of view of knowing the *exact* state of each part involved, it is invaluable. This latter knowledge cannot be gained solely from indicator cards, since these only denote approximate conditions, and are, therefore, somewhat unreliable. To obtain positive information air testing is necessary. For instance, it is possible to take indicator cards showing full compression, correct fuel injection, correct opening and closing of inlet and exhaust valves, and yet, when the air test is carried out, it may be found that all the valves are leaking slightly, although not sufficiently to affect the card reading. While it may be argued that the leakage, therefore, is not worth considering since it does not materially affect the running of the engine, and that when it does manifest itself to any extent it will be at once apparent from the indicator cards and operating symptoms, nevertheless it means that the true state of affairs is not known, and that by waiting until the symptoms develop may mean that when the part is opened up for overhaul the defect may be found to have extended to such an extent that a part renewal is necessary, thus adding to maintenance costs whereas the expenditure could probably have been avoided.

It is scarcely likely that a ship will have a single run of sufficient duration to wear out any part of an engine, but, before starting on a trip of any length it is most advisable to prepare against contingencies, and this can best be done by testing all parts as stated.

In addition to any work which may be noted during the voyage as having to be done on arrival in port, it is advisable to examine and clean all the main air compressor valves. Unfortunately, it is not possible to obtain accurate external indication as to the exact condition of these valves, with the consequence that when carbonization does manifest itself, the trouble will increase rapidly, necessitating the removal of the valves for cleaning, possibly before the ship's destination is reached. In addition to this, there is again the argument, that, by allowing the valves to operate too long, it may be necessary to renew parts, of which springs are the least consideration, whereas with earlier attention such replacements would not have been necessary.

Watch Keeping.—At the outset it may be said that watch keeping on a motor-ship is much easier than on a steamship, irrespective of size. Even with the largest present-day motorship installation, one engineer can comfortably superintend the running of all the plant in the E.R. during normal operation at sea when once the vessel has cleared the coast. Of course, in the bigger installations it is customary to have at least two engineers on watch at a time, but usually only the senior engineer does actual watch keeping, while the junior employs his time in overhauling auxiliary machinery and spare gear. By following such routine, "field-days" are eliminated, while the work of inspection and overhaul is kept up to date.

Before entering the E.R. to go on watch, it is advisable to have a look at the funnel exhausts and to note if any of the engines is smoking. With the modern air-injection Diesel engine, there is no excuse for a smoky exhaust, provided the load is normal with all parts in good condition and properly adjusted.

On reaching the top platform level, the points to observe are:—

(*a*) Temperature of cylinder head cooling water discharge. In the event of abnormal temperature being met with the engineer should not rest content with merely re-adjusting it, but should try to locate the cause. When he comes to read the piston-cooling water discharge temperatures, he should note how the figures compare with the cylinder head readings and draw an inference.

(*b*) That all valves are apparently functioning properly. In the case of inlet and exhaust valves attention should be paid to see that they seat properly, this being noted by the vibration made as the valve closes, and can be checked by feeling the valve spring cap for shock. An alteration in the tappet clearance might tend to prop the valve open, but only to a small degree, so that the effect might scarcely be noticed and yet the valve seat would be damaged by scoring, due to the action of the high temperature gases. Inlet valves suffering from this defect would evidence it by the sound of spitting to be heard at the induction pipes, but exhaust valves give no indication by sound and cannot be checked for slight leakage until the engine is stopped. Should a slight leakage develop to greater magnitude, it will begin to record by an increased pyrometer reading.

Should a valve spring be broken the valve would show irregular movement, and this is easily checked by comparison with the movement of the other valves. In the event of a spring completely breaking up (more common with high-speed engines) there would be no doubt as to what had gone wrong since the hammering noise would very soon draw attention to the defect.

Irregular movement can also be caused by the valve spindle tending to stick in its guide, but this defect is, usually, distinct, in that the valve shows more tendency to stay open as opposed to the shivering action caused by a broken spring. Of course, should the sticking develop to the extent of seizure, indication would be given either by the noise caused through the piston striking the open valve, in the case of an engine where the clearance space is small, or by the cylinder cutting out due to loss of compression.

(*c*) That the fuel injection valve packing glands are tight, and that no escape of fuel or blast air is taking place from any of the joints. The roller clearance of these valves should also be observed to ensure that it is neither excessively small or great. This clearance can only be checked within fine limits by taking a set of indicator cards, since it is possible to be experiencing late injection without any external indication whatever. It is only when the opening point is excessively late that it will show in the exhaust, or when excessively early, by the noise caused through detonation in the cylinder.

(*d*) (Where the top-level camshaft is used).

Camshaft journals and the like require no comment since they constitute everyday practice. Where the cams are concerned, the only one drawing attention is that for the fuel injection valve, since it almost invariably is made with an adjustable toe-piece, which, with the perversity of all things not intended to slacken back, may work loose.

Cam rollers want watching to ensure freedom of operation and apparently correct clearance being maintained.

In the case of a "Beardmore-Tosi" engine in which reversing is carried out by moving the rollers axially on their fulcrum pins, care must be taken to keep all the fulcrum pins clean and cam rollers particularly free.

Again, in the case of a "Beardmore-Tosi" engine which uses a director valve in conjunction with the combined inlet and exhaust valve, attention must be paid to the eccentrics operating the former, as indication is given of the condition of the director valves by observing the action of the eccentrics, much in the same way as the condition of the slide valve in a steam engine could be denoted by its operating eccentrics.

Then, the camshaft driving gears; these naturally would be inspected to see that the lubricating oil is reaching this, its highest level.

MARINE DIESEL ENGINES—RUNNING CONDITIONS AND OPERATING SYMPTOMS.

Middle Platform.—On passing to the middle platform the most important points are, in the case of a top-level camshaft engine, the bottom cylinder covers and piston-cooling gear glands. In the case of a bottom-level camshaft engine, such as a Burmeister and Wain, there must be added the camshaft, cam rollers, lay shaft and reversing gear, all parts needing same attention as in the case of top-level camshaft engine.

Each bottom cover should be inspected to see what amount of carbon deposit is on it, and the condition of this deposit should be noted. Under no circumstances should the deposit be allowed to accumulate to any appreciable extent, as there is always a danger of it working its way down through the piston-cooling gear glands and contaminating the water, finally causing undue wear of the gear including the piston-cooling pumps. Then there is also a likelihood of the deposit passing through the piston-rod glands and so contaminating the lubricating oil in the crankcase, finally manifesting itself in bearing trouble.

Yet another effect of too much deposit left lying on the bottom cylinder covers, is that it may be drawn up with the pistons and cause scoring of the cylinder liners. This mainly happens through the compression effect caused by the piston on its downward stroke, splashing the deposit asunder and causing bits to enter the space between the piston and liner walls.

When the bottom cylinder covers are being inspected it will be noted if there is any sound of gas escaping past the piston rings, or if there is a knock in the cylinder, only audible from this position.

It is important to note when piston-ring leakage first commences, as if all the symptoms at the start are noted and carefully analysed it may be possible to state decisively actually what caused the leakage. In this connection, the deposit on the bottom cylinder covers should be examined to determine, if possible, the conditions of lubrication within the cylinder.

Lubricating oil consumption is an important item in a motorship, and since the quantity used in the engine cylinders leaves no recoverable excess, naturally, it is desirable to reduce the amount used to a minimum. When the bottom cylinder covers show a pool of "liquid" oil lying, obviously the quantity being used is in excess of the amount required, but, on the other hand, if the deposit is too dry in appearance and is in the form of hard carbon or coke, although to all other indications the lubrication is sufficient, it would be well to increase the amount given until the deposit shows just a trace of softness.

Other conditions governing this question include the temperature of piston-cooling discharge water, since if this is allowed to rise abnormally at any time the cylinder lubrication will be affected. The cylinder head temperature bears on the question as well, as also does the exact nature of the combustion, these elements affecting the temperatures involved throughout the cycle and so affecting the cylinder lubrication to no small degree. Indeed, it is in this connection that conditions such as late injection are most pronounced in their effect, although it is a point usually ignored. After all, the lubrication of a cylinder should be varied according to the exact nature of combustion prevailing at the time instead of being left at a constant quantity, probably adjusted at a time when the engine was tuned up to concert pitch and under the best conditions.

In connection with piston cooling gear glands, also viewed from the bottom cylinder cover, it is important that the first sign of leakage from them should be noted, as this also bears directly on the question of cylinder lubrication and piston-ring tightness. If piston-cooling water leakage develops at this point, it is probable that it will find its way on to the piston and cylinder walls, with the result that when the pistons are withdrawn for survey or examination, it will be found that most of the rings are immovably stuck in the grooves. Water affects the piston rings and sets up corrosion, notwithstanding the presence of the oil and carbon with which the piston and rings are coated.

Referring to a knock which may only be heard from the position of the bottom cylinder cover, it is well to investigate the source at once. A light noise, more in the form of a click, will be made if a piston ring is broken and will occur towards the top and bottom of the stroke as the crank approaches and turns over the centre. Then it is possible to get a very light combustion knock which will be almost or entirely inaudible at the cylinder head due to the noise made by the valve gear. This is located easily since it can only occur at or about top centre during the firing stroke and can be confirmed by taking an indicator card, when the cause will be seen and can be remedied.

MARINE DIESEL ENGINES—RUNNING CONDITIONS AND OPERATING SYMPTOMS.

In passing along the middle platform it is advisable to occasionally feel the temperature of the starting air pipes where they join the cylinder heads. Any increase in temperature means that a starting air valve is leaking and that the combustion gases are passing back along the pipe. Of course, should the leakage be excessive, other indications would exist in addition to this, but it is better to make the discovery before much damage is done. A slight leakage can, as a rule, be cured merely by turning the valve on its seat a few times.

When it is an air injection engine with the air compressor driven off the main engine, it is usual to find at least the H.P. aftercooler on the middle platform level, while in all probability the I.P. and L.P. inlet and discharge pipes will be accessible from there as well. In connection with these, close observation is necessary if the best results are to accrue. It is unusual to find the compressor stage temperatures being read and recorded by thermometers when a ship is at sea in normal operation, but it is a practice that might well be inaugurated resulting in the ultimate reduction of maintenance costs. As matters stand at present, it is possble that most harm is caused to valves and springs by the unbalanced state of the stages when operating at nominal output, resulting in increased temperatures being encountered which finally carbonise the lubricating oil on the valves and remove the temper from the springs.

Bottom Platform.—On this level the main interest centres around the starting platform with its component gauges and recorders. Of these, reference may be made to the following points:—

Lubricating Oil Gauge.—The pressure, usually controlled at the lubricating oil pump, only denotes the force of the oil before entering the main bearings, so that observation of the highest point of delivery in the lubricating oil system is necessary to ensure adequate distribution taking place throughout. The lubricating oil pressure is mainly governed by the condition of the main bearings, to which it is first delivered from the pump, so that it would be quite possible to have considerable oil loss from bottom and top-end bushes, resulting in an inadequate supply going to the guides, etc., and yet the gauge pressure would be about the same figure.

A secondary gauge is usually fitted at the pump, and, in the case where the filters are arranged between the pump and the engine, comparison of the two gauges will denote the condition of the filter. Where the filter is arranged on the suction side of the pump, the condition of it can be ascertained by noting what additional amount of opening of the S.L. suction valve is necessary to maintain the desired pressure.

Cylinder Jacket Cooling Water Gauge.—This pressure is affected by control between the circulating pump and the inlet pipe to the jackets, and, once this is set, by readjustment of the discharge valves on the cylinder heads. It is unlikely to be affected by any untoward happening such as slight leakage from the combustion side into the water space, this, more probably, being detected by inspection of the overboard discharge. Silting in the water passages tending to choke them would ultimately show on the gauge, but this trouble would be more readily detected by signs of overheating which would become manifest ere a pressure difference was noticed.

Piston Cooling Water Gauge.—Pressure control is effected on the suction side of the piston cooling pump and by regulating valves at the inlet to each piston. Variation is attributable to the filters, the same indications existing as in the case of the lubricating oil pump, since a secondary gauge is also fitted to the piston cooling pump.

Since the resistance pressure depends entirely on the regulating valves at the inlets to the pistons, the engine pressure gauge is only arbitrary and will not record defects in the system other than the state of the filters. Any defect arising, such as a cracked piston, would only be detected by inspection of the piston cooling water return pipes, which, in this case, would show traces of escaping gases. Of course, other indications of this mishap would at once evidence themselves if the crack were of any magnitude, but a slight leakage might first be detected from the source of the piston cooling return water.

Blast Air Gauge.—Assuming the main engines to be running at normal output with the air compressor suction regulated, then control of the blast air pressure is mainly affected by adjusting the blow-off to the starting-air reservoirs, and, when these are fully charged, by allowing the surplus to escape to the atmosphere. Variation of blast air pressure, apart from the blow-off, may be due to air compressor faults, in which case verification can be had by reference to the air compressor gauges, or to defects arising in the fuel injection valves. A gradual, but persistent increase in pressure would be accounted for by a choked atomizer in one of the fuel valves, or might be due to an alteration occurring in the roller clearance of one of the same valves. Either case would probably be accompanied by a falling off in engine speed when the cause could be verified, first by examination along the camshaft to ensure that all clearances were apparently correct, and then by taking a set of indicator cards to check the conditions prevailing. If rough weather was being experienced at the time, it is possible that a drop in engine speed might be attributed to it, so that the blast pressure would form the best indication under the circumstances.

A decrease in blast pressure, necessitating reduction of the quantity of air blown off, may be due, apart from compressor troubles, to a leakage having developed at one of the various joints in the pipe line or at a fuel injection valve itself. The most usual source of loss is due to leaking packing glands in the fuel injection valves, but such packing glands are already becoming things of the past.

A leaking fuel valve will cause a gradual drop in the blast pressure, but, when it extends to any magnitude, the effect will become pronounced and is likely to be traced from other sources.

A fuel injection valve that is leaking badly causes pre-ignition due to a higher compression pressure, formed by the blast air entering the cylinder continuously, thus giving a higher initial pressure at the start of the compression stroke; also by the fuel being blown in as soon as it is delivered from the pump to the injection valve. The defect will also be recorded by traces of smoke in the exhaust.

Pyrometers.—These consist of thermo-couples placed as close as possible to the exhaust valves in each cylinder, while the wires are led into an interpole switch and the readings are taken from a common dial.

Pyrometers are invaluable for denoting the condition of the exhaust valves, if used in conjunction with indicator cards, while the balance of the engine can be approximated also by their use, provided the condition of the valves is known. For instance, if the exhaust temperature of any one cylinder was observed to be lower than normal, it would probably be found that the cylinder was low in output due to fuel admission, or loss of compression due to piston rings or sources other than the exhaust valve. If the temperature were above normal, then the cylinder would be overloaded or fuel injection be late, or else the exhaust valve would be leaking. With a leaking exhaust valve a higher pyrometer temperature would be recorded even though the fuel distribution to that cylinder was less than its neighbours. This, of course, is due to the escape of gases right through the firing stroke. Therefore, if all valves, piston rings, and cylinder joints were known to be tight, uneven pyrometer readings would draw attention to fuel injection and load balancing conditions.

Air Compressors.—Some idea of the operating symptoms of an air compressor of a type shown diagrammatically in the accompanying figure, can be gathered from the following notes:—

Valves.

(1) Should the L.P. suction valve be choked the L.P. suction gauge will record a vacuum, accompanied by a drop in all stage pressures.

(2) Should the L.P. suction valve be leaking, indication will be given by the noise of blow-back at the air intake.

(3) Should the L.P. delivery valve be choked, the L.P. cylinder will experience a higher terminal pressure, but the L.P. gauge will record a lower pressure than normal, since the gauge is connected on the delivery side. The effects on the compressor will be to give a high L.P. delivery temperature accompanied by a low I.P. gauge pressure and reduced output.

(4) Should the L.P. delivery valve be leaking, the defect will show on the L.P. gauge, since during the compression stroke the pressure will rise, while on the suction stroke the pressure will fall rapidly due to some of the remaining air in the L.P. intercooler passing back into the L.P. cylinder. As the leakage develops, or should the valve break, the L.P. terminal pressure will drop, since the air passing back into the cylinder on the suction stroke will prevent the L.P. suction valve from opening properly. The I.P. gauge pressure will also drop while the H.P. delivery temperature will rise.

(5) Should the I.P. suction valve be choked, the L.P. gauge pressure will rise due to the increased resistance, while the I.P. gauge pressure will drop and the output of the compressor be reduced.

(6) Should the I.P. suction valve be leaking the L.P. gauge pressure will rise since the L.P. stage is doing normal work,

while the I.P. stage is not delivering its full output during the compression stroke. the I.P. gauge pressure will fall and the compressor output will be reduced.

(7) Should the I.P. delivery valve be choked, the I.P. gauge pressure will fall due to lack of supply, while the L.P. gauge pressure will rise due to the increased resistance.

(8) Should the I.P. delivery valve be leaking the I.P. gauge pressure will fall due to the air from the I.P. intercooler enter-

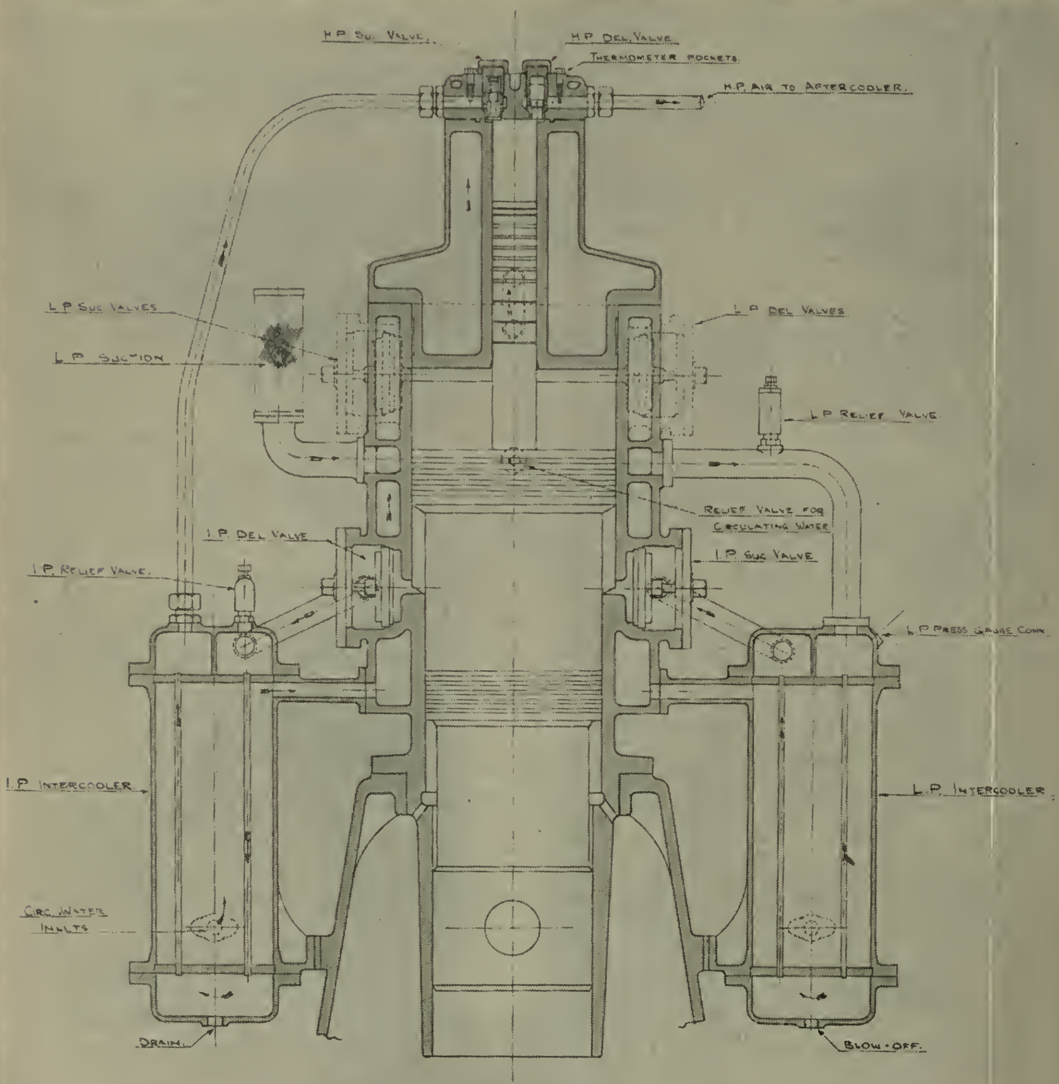


Fig. 1.
Air Compressor.

ing the cylinder on the suction stroke, thus reducing the amount taken from the L.P. intercooler, with the result that the L.P. gauge pressure will rise as the machine is tending to become a two-stage compressor. So long as the intercooler relief valves do not lift the output will remain unaltered, although all temperatures will rise considerably.

(9) Should the H.P. suction valve be choked, the I.P. gauge pressure will rise due to increased resistance, while the L.P. pressure will gradually rise owing to less air being drawn by the I.P. cylinder, since the amount of air in the clearance space of the I.P. cylinder at the end of its compressor stroke is at a higher pressure, and will, accordingly, expand to greater volume during the suction stroke. This will limit the amount of air passed from the L.P. stage, with the result that the L.P. pressure will also rise.

(10) Should the H.P. suction valve leak or break, the I.P. gauge pressure will rise rapidly accompanied by a rise in the L.P. gauge pressure, since the machine would again tend to become a two-stage compressor, this time with the I.P. cylinder acting as H.P. stage. The relief valves would, of course, prevent this actually happening.

(11) Should the H.P. delivery valve be choked, the I.P. and L.P. gauge pressures will rise gradually, due to the increased pressure at the end of the compression stroke of the H.P. cylinder, with its ensuing expansion to greater volume preventing the correct volume of air passing from the other stages, and so increasing the pressure in them. The output of the compressor would also drop.

(12) Should the H.P. delivery valve leak or break, the machine would tend to become a two-stage compressor just the same as the case of a broken H.P. suction valve.

Piston Rings.

(1) Should the L.P. piston rings be defective, then on the L.P. compression stroke some of the air would get past them into the I.P. cylinder, although this would not matter. On the L.P. suction stroke, however, which is also the I.P. compression stroke, some of the I.P. air would escape back to the L.P. cylinder and so lower the volumetric efficiency with a consequent drop in output. If the leakage were only slight, the L.P. pressure would remain about normal or slightly higher, but the I.P. pressure would drop.

(2) I.P. piston-ring leakage would mean the escape of air to the crankcase with a reduced I.P. gauge pressure.

(3) Should the H.P. piston rings leak, the L.P. gauge pressure would rise and the output of the machine would drop considerably. If such leakage develops to any extent the L.P. relief valve will blow off intermittently. It will be found that the L.P. gauge pressure rises excessively on each compression stroke until the relief valve opens, when the pressure will instantly drop to considerably less than normal, while several strokes will be required to again raise it to the blowing-off point.

Other Defects.—Air leakage into the Water Space.

The effect of this is to interrupt the circulation and cause overheating, and is usually detected by violent oscillation of the compressor cooling water gauge pointer. The source of leakage can usually be located by a rise in jacket water temperature, the heating being evident from the source upwards, since the effect of the leakage is the same as if a baffle plate were inserted at the same place, *e.g.*, if the leakage were taking place in the H.P. stage, the jacket temperature would rise at that point, while at the other stages the temperature would remain normal. Likewise, were the leakage to take place in the L.P. intercooler, the temperature would rise from there and show in the I.P. and H.P. stages as well, while the L.P. cylinder jacket temperature would remain normal.

A very slight leakage would be hard to detect when running, but if one is suspected through indication given by the vent cock fitted at the highest point in the system, it can best be located when the engine is stopped. This can best be done, either by applying water pressure to the jackets and watching for leakage in the air spaces, first having removed all the valves and opened the intercooler air draincocks, or by applying air pressure to the stages and intercoolers, and, having removed some of the water service pipes, by listening at the ports, when the leakage can be detected by sound. The latter method is the better, having the advantage of higher test pressure.

Conditions of Intercoolers and Aftercoolers.—Some idea of the state of the above can be formed under running conditions. If the air inlet temperature to any intercooler is normal and the discharge temperature is high, then, if the jacket temperature is normal, it indicates that the tubes are dirty. This imperfect cooling may be caused by excessive lubrication, since an oil film in the tubes lowers the thermal conductivity. Again, should the air inlet temperature be normal and the discharge temperature show lower than normal, having regard to jacket water temperature, this indicates that the tubes are wearing thin, erosion having taken place. In this connection, care should be taken to keep all intercoolers well drained and to see that the amount of moisture present is not excessive. When a compressor is working correctly, the water from the drains should have the appearance of milk with an occasional spot of oil present in the flow.

From the foregoing it will be seen that the various defects have a degree of similarity in their symptoms that make careful reasoning necessary to accurately diagnose the source of trouble. It will also be seen that reference to the stage temperatures is of considerable assistance, while it might be mentioned that the use of a stethoscope is invaluable for locating defects, particularly where valves are concerned.

As was mentioned earlier in the paper, even if the compressor is operating satisfactorily, it is not advisable to wait until indication of choked valves is given, as with the generous area usually allowed in the design of these valves, quite a considerable amount of carbonisation can take place before any external indication will exist. What has to be determined is the time that can be safely allowed to elapse before withdrawing the valves for cleaning. This will depend on the grade and quantity of lubricating oil used and the temperature balancing of the stages, and can only be found by trial.

Starting and Manœuvring.—The great problem under this heading is conservation of blast and starting air.

Referring to Fig. 2, which shows diagrammatically a six-cylinder four-stroke engine, it will be seen that such an engine has, regardless of position, always got at least—

(*a*) One cylinder on the firing or expansion stroke.

(*b*) One cylinder on the compression or firing stroke.

(*c*) One cylinder on the suction or compression stroke.

Now, even under the worst condition, which is that shown in the illustration, all the air that is needed to start the engine is for one impulse to be given to cylinder No. 1, followed by an impulse to cylinder No. 5, since the piston in cylinder No. 3 is rising on the compression stroke and will receive a charge of fuel about top centre which will effectively keep the engine in motion until the next fuel valve opens in sequence. Should the fuel charge given to cylinder No. 3 fail to ignite, then, assuming that all air-starting valves have been cut out, another air impulse must be given, this time to cylinder No. 6, which will bring the piston in cylinder No. 2 to top centre, where it will receive a fuel charge. There is nothing to be gained by allowing the engine to make several revolutions with all starting valves in action. On the contrary, the effect is to cool all the cylinders and to introduce moisture, which passes with the starting air, and which acts as a great deterrent to the ignition of the fuel due to the heat absorbed in its evaporation.

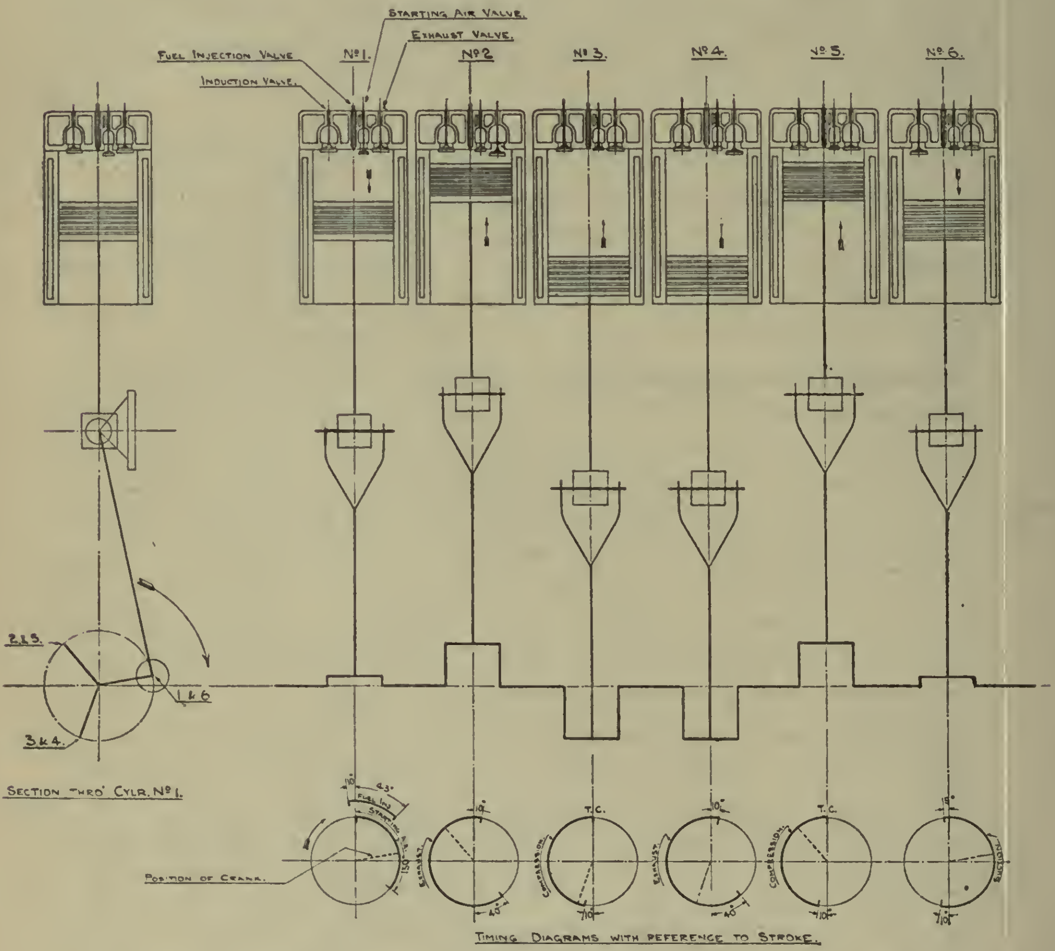


Fig. 2.

MARINE DIESEL ENGINES—RUNNING CONDITIONS AND OPERATING SYMPTOMS.

Provided the cylinder head valves and piston-rings are in a reasonable state, the starting acceleration due to one air impulse is sufficient to raise the temperature on a single compression stroke, to a degree high enough to ensure ignition of the fuel charge. It has been the author's experience that an air pressure merely high enough to turn the engine without sticking is sufficient to ensure a start on fuel, and when it is considered that the pressure referred to is but 180 lbs. per sq. in., it will be gathered that the acceleration rate is not very high.

When an engine does not readily pick-up on fuel, the most common reason is due to air in the fuel piping between the injection pumps and injection valves. The greatest care should, therefore, be taken to ensure that the process of "flooding through" is carried out thoroughly, as on this, more than anything else, instantaneous fuel starting depends.

It will be readily understood that the most economic consumption of starting air depends upon the period elapsing between the time the starting valve is cut out and the fuel valve is brought into operation, so that the time element plays an important part.

Fig. No. 3 shows the operation sequence of the manœuvring gear for a Beardmore-Tosi engine. Four leading positions are given:—

(1) Six cylinders on air.
(2) Three cylinders on air; three cylinders on fuel.
(3) Six cylinders on fuel.
(4) Low injection conditions.

With this gear, position (1) is only provided to ensure the engine making a start, and when it does the gear is immediately moved to position (2) where the first fuel injection valve comes into action. If the firing is instantaneous the gear is moved to position (3) thus cutting out the remaining starting valves and at the same time bringing into action the remaining three fuel injection valves. After the engine has made a few revolutions with all fuel valves in operation, there is no longer any need to pause with the manœuvring gear at any intermediate position, but, instead, it can be run right from the neutral to position (3) each time. Although the manœuvring gear is kept in motion right through the sequence, the design is such that a short air impulse given to one cylinder as the gear passes position (1) while at position (2) fuel injection takes place in one of three cylinders, accompanied by another short air impulse in one of the remaining three cylinders. In this way the engine gradually gathers speed until position (3) is reached, by which time all six fuel injection valves have come into operation and are in the position of maximum opening.

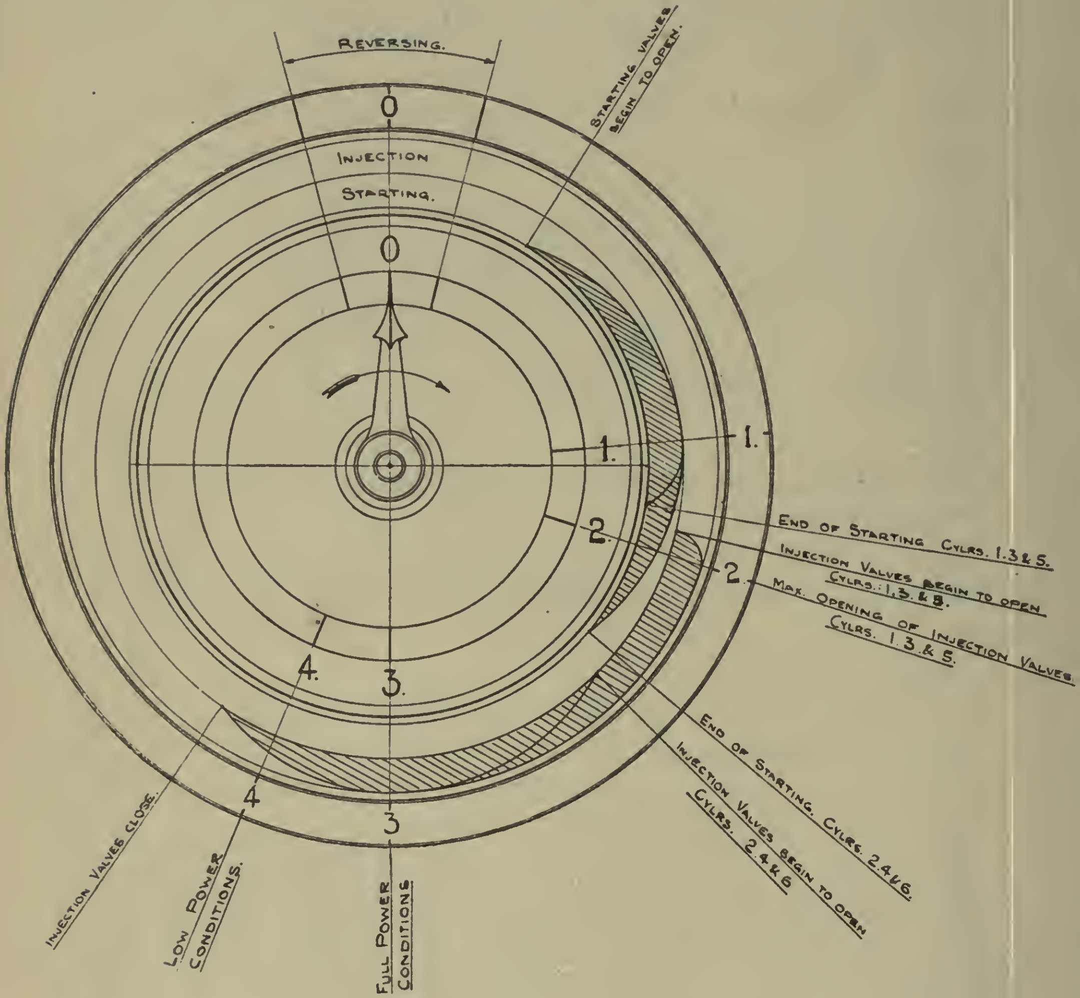

Reversing.
0
Injection
Starting.
0
Starting Valves Begin to Open.
1.
1.
End of Starting Cylrs. 1.3 & 5.
2.
2.
Injection Valves Begin to Open Cylrs.: 1.3. & 5.
Max. Opening of Injection Valves Cylrs. 1.3. & 5.
End of Starting. Cylrs. 2.4 & 6.
Injection Valves Begin to Open Cylrs. 2.4 & 6
4.
3.
4
3
Injection Valves Close.
Low Power Conditions.
Full Power Conditions

Fig. 3.

Sequence of Manoeuvring Operations.

The amount of starting-air used in giving the short impulse is actually less than the quantity which would be used in one starting valve alone, were the gear stopped at positions (1) or (2).

Considering now the blast air question. The principal saving in blast air is effected by manœuvring the engines with the lowest possible pressure, in cases where the fuel injection valve lift remains a constant, and by reducing the fuel valve lift and also the blast pressure in cases where variable injection valve timing is provided. For the best manœuvring conditions, as well as the greatest flexibility, a variable gear is essential.

When manœuvring an engine it is considered desirable to empty the blast air pipes between the control valve and the fuel injection valves as soon as the engine is stopped, since there is a danger of any air that remains in the fuel injection valves or blast pipings finding its way past the fuel non-return valves into the fuel piping and so causing an air lock, which might even extend to the fuel injection pumps. It is this source of loss that constitutes the greatest consumption of blast air, since the pipe line and fuel valves have to be refilled each time a start is made. The author is rather of the opinion that this emptying of the system should be abolished, and that more trust should be placed in the efficacy of the fuel non-return valves. Experience with auxiliary engines, where draining of the blast piping is more the execption than the rule, has gone to show that an engine can be started after lying for a whole day with blast air in the system, without needing to be "flooded through" again.

Apart from the loss incurred through draining, the consumption of blast air is not at all high when manœuvring, especially where variable valve gear is fitted. By "linking-in" to use a steam engine expression, or, in other words, by reducing the lift of the fuel valves, a considerable saving of air is effected. Again referring to Fig. 3, it will be seen from the diagram that the fuel valve lift can be varied from a maximum at position (3) to a minimum at position (4) after which the valves cut-out altogether. By varying the gear between positions (3) and (4) the most suitable lift and duration of fuel valve opening can be selected to suit the particular load, so that the engine will run with an invisible exhaust at all speeds, and at the same time the consumption of blast air is the most economic possible. In going dead slow the gear is moved to position (4) and with the fuel regulated to suit, the engine speed can be reduced to 28 revolutions per minute. When the gear is at position (4) the engine will run for several minutes with the blast air shut off at the blast bottle, the air in the pipe line being sufficient for injection purposes.

MARINE DIESEL ENGINES—RUNNING CONDITIONS AND OPERATING SYMPTOMS.

With all engines, the aim is to make as much blast air as possible, to use the least quantity for injection purposes, and to pass the surplus to augment the starting air supply. Where a variable gear is not fitted, this is often done at the expense of good combustion and with dire results to the fuel injection valves in particular. The blast pressure is reduced to the lowest possible figure, about 560 lbs. per square inch, and while combustion remains normally at constant pressure all is well, but should it change to approximately constant volume with an accompanying pressure rise, then there is the danger of an explosion taking place in the fuel valve, Another effect is, that part of the products of combustion are forced back into the atomizer space and tend to choke the slots or holes. This choking is not noticed at the time, unfortunately, but as soon as " full away " is received it is found that the defective valve will not pass the necessary quantity of oil in the given time, thus the speed falls due to late injection. In cases where the choking is great, the engine has to be stopped and the valve changed.

Where variable valve gear is fitted, an important point to observe is that the fuel valve lift is at all times strictly proportioned to the quantity of fuel being delivered by the pumps. In executing telegraph orders for slow speed, the necessary reduction in revolutions is often got by merely " linking-in " without first reducing the amount of fuel being supplied by the pumps. The effect is, that the oil, being supplied at a greater rate than the reduced lift can cope with, builds up in the air space of the fuel valve. Should the slow speed order prove to be of long duration, the fuel will find its way into the blast air piping where its presence is dangerous. Again, should a demand for increased speed be received and the valve lift increased accordingly, a violent explosion in the cylinder will result due to the excess oil passing in bulk. Even in the first case of operating at slow speed, detonation will occasionally occur due to this same cause. The most probable reason is that the fuel tends to enter the cylinder in a solid state instead of being atomized—the depth of oil lying in the valve becoming too great for the blast air to properly penetrate it, so that atomization is impossible.

The most advisable procedure when operating an engine with very reduced fuel valve lift, is to occasionally back the gear into mid position and so allow the fuel valve to clear itself. In doing this the revolution speed will rise, but if the gear is immediately returned to its original position the momentary increase will prove negligible.

Another point, although bearing indirectly on the question of conservation of air, is, nevertheless, an important one; that is, starting momentum. Referring to Fig. 4a, two starting curves are shown, the lower one being the ideal.

When an engine is being started in answer to a telegraph order for slow speed, it is quite usual to see the revolution speed rising momentarily to an abnormal degree as shown by the upper curve in Fig. 4a, and then, when all fuel valves are in operation and the engine has settled down, the speed dropping to a steady figure, say 40 revs. per min. The reason for this irregular acceleration is due to some fuel having been trapped in the injection valves as the manœuvring gear was run into neutral when previously stopping the engine. This fuel, plus an additional quantity supplied by the pumps when starting, is injected into the cylinders as the manœuvring gear passes from positions (2) to (3) with the result that the revolutions rise abnormally until position (4) is reached, when the speed drops due to the effect of reduced fuel valve lift. The effect of this is to give a powerful impulse to the ship, with the result that the order is curtailed to a very short duration, should the ship be docking at the time. Since it is during docking operations that the most frequent demands for engine movements are received, and, therefore, the greatest need for the economical use of air exists, it is highly desirable that the starting momentum should be

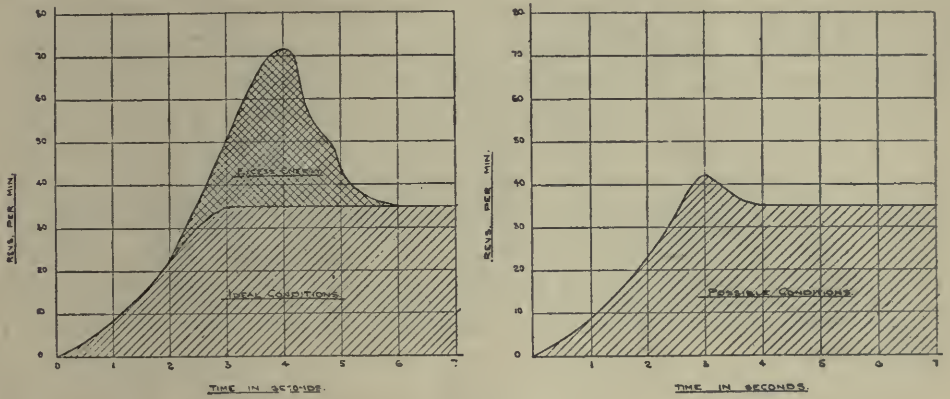


Fig. 4a.

Fig. 4b.

Curves Showing Starting Momentum.

reduced to a minimum so as to prolong each movement to maximum duration, thus providing an opportunity for the engine to make up some of the air that has been used in starting.

In order to make an engine start with a momentum more nearly approaching the ideal, as shown by the lower curve in Fig. 4a, the following procedure is necessary.

Assuming the engine to be running, when the order to stop is received, the fuel control lever should be moved into the "cut-out" position, while the manœuvring gear should be put in position (4) (see Fig. 3) and kept there until the engine stops, when the gear can be returned to neutral. Doing this ensures the injection valves being thoroughly cleared of fuel, while the amount of blast-air used during this manœuvre is negligible, and the time element is slight and need scarcely be considered. Upon re-starting, the fuel control lever is placed in the position approximating to slow speed and the manœuvring gear is then put through its normal sequence to position (4) when it will be found that the engine starts in a manner approaching the curve (Fig. 4b). The slight rise in speed that still exists is probably due to combustion taking place at approximately constant volume, this being caused by the injection valve lift being at a maximum when the small quantity of fuel first enters the cylinders.

At this juncture it is interesting to consider the question of the most advantageous position for the control platform on a Diesel engine.

In the majority of cases the control gear is arranged on the bottom platform level, following standard steam practice and tradition, but the arrangement is open to criticism where top-level camshaft engines are concerned. When this type of engine is specially considered, it will be found that most of the gear deserving the closest attention is arranged at the top of the engine, while at the control platform attention is confined mainly to gauges. Were the controls arranged at the top-platform level, then all the valve gear would be under more constant supervision, since it is on the control platform that most of the time on watch is, of necessity, spent. Again, the fuel settling tanks are almost invariably arranged on the top platform, and these would be more readily checked and handled when filling. Another advantage lies in the fact that it is possible to obtain a more comprehensive grasp of the action of the engine when manœuvring, which can be made to result in a saving of starting air—a very important point.

If it is argued that the top platform is not neglected with the present arrangement, then it can be equally assumed, that were the controls arranged on top, the rest of the engine would not be neglected.

While auxiliary engines are usually arranged on the bottom platform, this would not matter, since, even with the present arrangement, they are not under constant supervision, while the same indication of their running, apart from actual examination, exists equally well at the top platform level.

It will be found that in many of the modern double-reduction steam turbine engine rooms, the control platform is raised to the turbine level, while the air, feed, and circulating pumps are arranged under it and out of sight. Even the dynamo sets, in cases where most of the auxiliaries are electrically driven, are also far removed from the control platform, yet the arrangement would appear to be entirely satisfactory.

What applies to the steam engine room as quoted, is equally applicable to the engine-room of a motor ship, while the question resolves itself into placing the control platform, if at all possible, at the place where most supervision is required. For a Diesel engine with top-level camshaft, the top platform is undoubtedly that place.

Auxiliary Engines.

Much of what has been stated regarding main engines is applicable to auxiliary engines, but there are other factors preventing the application as a whole.

Since there is at least always one auxiliary engine as stand-by, maintenance work is rendered easier, while the engines can always be kept in a state to allow of full output being developed when required.

Air testing of the cylinders and valves is even more necessary in the case of auxiliary engines if the exact condition of each part is to be known, as indicator cards taken from these engines are even less reliable than main engine cards, owing to the higher speed and greater inertia effects of the indicator.

Again, auxiliary engines operate under rather more adverse conditions than the main engines, this being caused by varying loads and the difficulty of maintaining perfect combustion under each. In view of this, it is advisable only to run the engines for a comparatively short time—say four days at a stretch—until the service time for each constituent part is established.

The worst conditions in respect of combustion, are met with when loading or discharging cargo, and, at the best, a compromise has to be effected to ensure reasonable operation throughout the range of loads.

When the load is light, the blast pressure should be 600 lbs. per sq. inch or under (dependent upon the compression pressures) while the fuel injection valve lift should be as small as possible, these conditions tending to maintain steady firing with combustion at constant pressure.

Heavy loads, on the contrary, require a high blast pressure and the fuel injection valve lift increased if the best results are to be attained, while anything in the nature of an overload necessitates a blast pressure of at least 1,000 lbs. per sq. inch, in conjunction with the maximum fuel injection valve lift. Since cargo work involves loads ranging from zero to 120% neither extreme in adjustment is suitable, so that settings have to be arranged which will allow the engine to operate under reasonable conditions over the entire range. This is usually effected by keeping the blast pressure about 750 lbs. per sq. inch and by adjusting the fuel injection valve lift to 75% maximum. With these settings the engine runs best at moderate load, but will take full load for short periods, while on light load continuity of firing is good provided the period is not too long. Of course, a variation of these settings is advisable according to the average load experienced. On some occasions the average load might be to the light side, when reduced blast pressure and fuel lift valve would be desirable, while, on the other hand, a heavier average load would best be met with an increase of blast pressure and fuel valve lift. The question becomes more important when two or more engines are run in parallel, since irregular running means uneven loading, and may end in one engine cutting out when the load goes off momentarily, so that the remaining engine has to sustain the increase when the load again comes on.

The effect of high blast pressure, in conjunction with excessive fuel injection valve lift, during light loads, is to cause misfiring. The reason for this is due to the expansion of air through the fuel valve causing a reduction of temperature in the cylinder with the result that combustion is occasionally prevented, especially when the governor has reduced the quantity of fuel being delivered. The fuel, however, is partially vaporised and shows in the exhaust as white smoke, while the cylinder is never wholly scavenged. The result of this is, that over a stroke or two a combustible mixture is formed which finally explodes during a compression stroke, causing detonation and providing ignition for the following charge of fuel. This may again heat up the cylinder, so that firing is regular for a few strokes until the governor again reduces the quantity when misfiring once more becomes evident.

Low blast pressure and reduced fuel injection valve lift during heavy loads shows in the exhaust as black smoke, this being caused through imperfect atomization of the fuel, with the result that combustion is incomplete and the power is reduced. After-burning also takes place, which, in turn, reduces the weight of air taken in on the suction stroke, thus tending to still further lower the output.

Apart from thermal efficiency, the nature of combustion has its effect upon valves and piston-rings, and it is this effect the marine engineer is more concerned with. Misfiring results in carbonisation of piston rings, exhaust valves and ports, while incomplete combustion, involving after-burning, has its effect on exhaust valves in particular, the tendency being for these to burn at the faces.

Pyrometers are not usually fitted to auxiliary engines, and unless a suitable thermometer is available, the exhaust gas temperature cannot be utilised as a guide to the condition of the exhaust valves. For indication during running, therefore, it is necessary to depend upon indicator cards and the appearance of the exhaust gases, but this method has serious drawbacks, as already enumerated in connection with the main engines.

The operating symptoms of suction, fuel injection and starting valves, are the same as for the main engine and need no comment.

In connection with the lubrication of auxiliary engines, attention should be paid to the condition of the oil in the system as it is more liable to contamination than is the lubricating oil in the main engines. The reason for this is that auxiliary engines, being fitted with trunk pistons, all the burnt and carbonised oil passes direct into the crankcase. Again, unless the pistons are provided with extra long skirts, it will be found that a quantity of oil is splashed on to the underside of the piston crowns, and this oil also forms carbon which helps to contaminate the lubricating oil supply.

Cylinder lubrication is materially assisted by the amount of splash in the crank case, to such an extent, indeed, that the extra supply given by the mechanical lubricators is almost unnecessary. This is amply proved by air compressors fitted with trunk pistons open to a crankcase. Where these are concerned, the great difficulty is to prevent an excessive supply of lubricating oil passing into the compressor cylinder, and, if this is attributed to suction effect, then suction effect must also become manifest in the power cylinders.

Regarding jacket and cylinder head cooling water temperature, this is a point to which more attention might be paid with advantage. It is quite usual to find auxiliary engines running with the cylinder head discharge water at a very low temperature. This is principally due to the difficulty experienced in regulation caused by the supply pipes being of too generous dimensions, and the fact that it is cold sea water that is being used as a medium. Regulation, to attain a reasonable working discharge—say, 120 degs. F.—demands a lot of adjustment on the suction-side of the pump and the cylinder head regulating valves, but considerable improvement results, both thermally and mechanically. While running at sea, when the load is practically constant, adjustment is comparatively simple and re-acts principally in a saving of fuel, but when working cargo in port under varying load the adjustment becomes more difficult, although the results are equally valuable. By maintaining a discharge temperature of at least 100 degs. F., much steadier running over a wide range of loads is assured; while misfiring during light and no-load periods is greatly reduced, all of which reflects in the better condition of the valves and piston rings in addition to increased thermal efficiency. Temperature stress through the cylinder liners and heads is also lessened, and this is no small point if a heavy load has to be sustained for some time.

When working an auxiliary engine under the better thermal conditions, attention must be paid to the cooling water discharge regulating valves. As previously mentioned, due to the water pipes being too large in the bore, it will be found necessary to keep the discharge valves almost shut, and if left too long in this position, they will erode rapidly. It is advisable, therefore, to open them fully at regular intervals and so allow the deposits in the system to be thoroughly washed out. This can be done before shutting down, or afterwards, if the water supply is taken from an independent pump. It may be mentioned that this is an advisable procedure for main as well as auxiliary engines, and should be faithfully carried out each time before or after shutting down.

Considering now the air compressors fitted to auxiliary engines; these really have to play a dual part, in that not only has each to supply blast air to its own engine, but, when the main engines are being manœuvred, it is sometimes necessary to draw upon the surplus air from the auxiliary engine compressors. This necessitates these compressors always being capable of full output when entering or leaving port, and this can only be assured by cleaning the valves, ports and passages at regular intervals. What has been stated in connection with main engine compressors applies equally to those of the auxiliary engines. Where with the main engine compressors a certain amount of carbonisation of valves, etc., is unavoidable, due mainly to the time that must elapse before they can be cleaned, there is, therefore, all the more reason why the auxiliary engine compressors should be kept in the best condition.

Auxiliary engine compressor valves will tend to carbonise more readily than those of the main compressors, due mainly to more lubrication being given throughout, the unwanted supply coming from the crankcase splash and being unduly increased by higher revolution speed and reduced clearance space in the crankcase. Another reason is that higher compression temperatures are involved through the compressors being, usually, of the two-stage type.

Then stage balance also plays an important part. It is usual to find the compressors capable of giving at least 25% more output than is needed for blast during normal loads. The problem, then, is to operate the compressors at 75% output and still maintain a good temperature balance. The usual procedure would appear to be to regulate the quantity of air supplied by throttling the L.P. suction intake, or by blowing-off a certain quantity of air after it has been compressed in the L.P. stage. Neither system is good unless the H.P. stage is balanced to suit. To do this, it is necessary to work with a higher L.P. pressure, while the surplus air can be discharged through the H.P. drains. Although this system absorbs more power in effect, the cost is amply balanced by a reduced maintenance bill.

The best temperature balance is achieved by working each stage at the same compression ratio, having regard to the efficacy of the intercoolers, but although this is not always possible, adjustments can be made to bring the balance within reason. Compression temperature troubles incurred through wrong stage balancing can be practically abolished by the provision of suitable thermometers in the ship's outfit.

With regard to starting auxiliary engines, what has been said of the fallacy of running the main engines with all cylinders on air once the engine is in motion, applies equally to the auxiliaries. With them, however, it is usually necessary to give more than one air impulse before introducing the fuel, unless the starting air pressure is very high, as a greater momentum is necessary to ensure the compression temperature being sufficiently high. This is due to the fact that the smaller the cylinder the greater proportionately does the amount of cooling surface become, so that the minimum revolution speed is higher than in the case of a large engine.

The necessary speed for starting an auxiliary engine becomes a matter of judgment. If the engine starts briskly, one air impulse to each cylinder should be sufficient before introducing fuel, but even where starting is sluggish, due to low air pressure, it is advisable to cut out one starting valve after each cylinder has had an air impulse, and allow the engine to run with one cylinder on air and one on fuel until steady firing results.

It is sometimes advantageous, when starting with low starting air pressure, to allow the engine to make the first revolution or two with the blast air shut off, but with the fuel pumps in action, and then, as the first fuel injection valve is cut-in, to open the blast air valve. Doing this approximates more nearly to solid injection which has the advantage of readier combustion.

In a case where the blast air pressure as well as the starting air pressure is low, and it is not possible to recharge the reservoirs, the best method to adopt is to give a small injection of paraffin through each indicator cock before starting. The cylinder relief valves may lift, but the result is effective, although this method is not to be recommended as habitual practice every time an auxiliary is being started.

The trend in design of auxiliary engines appears to be towards a reduction in revolution speed, and it is a point that will be appreciated by those marine engineers who have sailed with auxiliary engines running at 300 revs. per min. and over. Unless in cases where expense has been no object, the reciprocating weights of such engines are usually excessive, resulting in great inertia load with consequent wear of bearings, especially top-half bushes. Such engines usually require all main and connecting-rod bushes to be overhauled and adjusted every six months, especially in cases where the lubricating oil has not been properly filtered and cleaned during that period.

In conclusion, it would be well to draw attention to the necessity for organised routine. No hard and fast rules can be laid down for the overhauling time of any parts, as it is all a matter of circumstance. Extra work will be essential at the start when dealing with new engines, until such time as definite information is established, but, as already pointed out, such work will be justified by a reduction of maintenance costs.

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